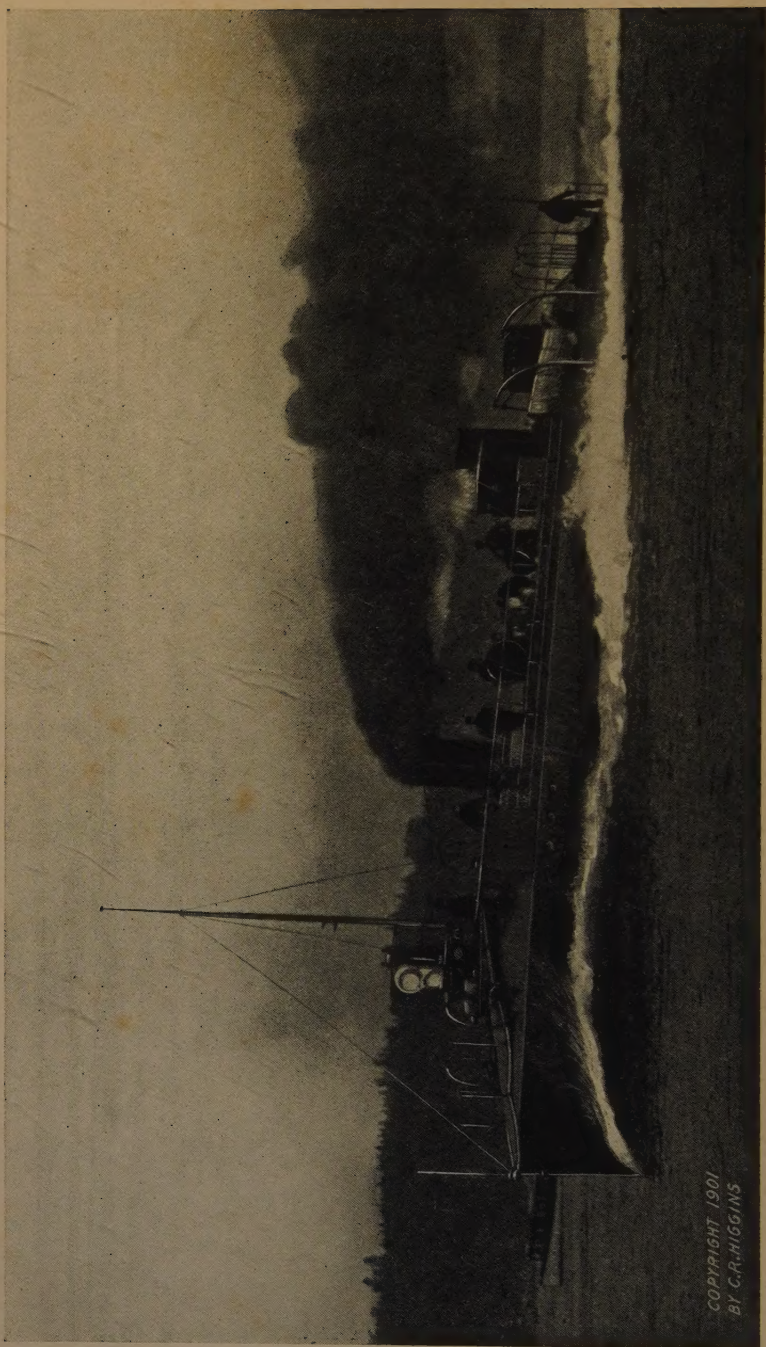


T. T. YOUNG FELTS

H. A. JOHNSON. JULY - 65

NAVAL RECIPROCATING ENGINES
AND
AUXILIARY MACHINERY

BARTON AND STICKNEY



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U. S. TORPEDO BOAT BIDDLE

NAVAL RECIPROCATING ENGINES
AND
AUXILIARY MACHINERY

A TEXT-BOOK FOR THE INSTRUCTION
OF MIDSHIPMEN

AT THE
U. S. NAVAL ACADEMY

BY
JOHN K. BARTON
Commander, U. S. Navy
Former Head of the Department of Marine Engineering
and Naval Construction

THIRD EDITION—REVISED AND REWRITTEN

BY
H. O. STICKNEY
Commander, U. S. Navy
Recently Head of the Department of Marine Engineering
and Naval Construction

ANNAPOLIS, MD.
THE UNITED STATES NAVAL INSTITUTE

1914

NAVAL RECIPROCATING ENGINES
AND
AUXILIARY MACHINERY

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PREFACE

The special aim of the writer in preparing this book was to adapt one to suit the conditions existing at the U. S. Naval Academy, where, with a limited time allowance, the student must be prepared to assume active engineering duties afloat as soon as practicable. With this end in view, a sound general knowledge only of such parts of the general field of engineering as is of special and practical use to the naval engineer has been attempted in the class rooms and laboratory, supplemented by a considerable amount of practical work in the various shops of the engineering building, together with the practical management of the engines and boilers of the various torpedo boats, destroyers, monitors, and a large fleet of steam launches attached to this station.

No mention has been made of the design of steam machinery, which is taught separately, the particular function of this book in the Department of Marine Engineering and Naval Construction being the construction, operation, management and care of machinery found on board naval vessels. The subject is a large one and to keep the matter within range it has been condensed as much as possible; especially is this the case with theoretic considerations, which it is to be hoped will be attended to in a post-graduate course.

As the Midshipmen are placed on duty immediately after graduation, details, descriptions and drawings selected for illustration are of machines in actual use in the U. S. Navy, such as will be encountered in any ship in our navy, and with a familiarity with the installation on board of a naval vessel no difficulty should be experienced in handling any other.

Use has been made of the various pamphlets prepared by instructors formerly on duty in this department, and in Chapter II the

notes of Lieut.-Commander F. C. Bieg, U. S. N., furnished valuable suggestions and assistance.

For Chapter XVI, on the important subject of "Engine Balancing," the writer is indebted to Professor I. N. Hollis, of Harvard University, Class of 1878, U. S. N. A.

Grateful acknowledgment is made to Rear-Admiral C. W. Rae, U. S. N., Chief of Bureau of Steam Engineering, and to his assistants, for the use of material available in the Bureau; to the manufacturers of the various auxiliaries used on board American naval vessels for the details and data illustrating their specialties, and to Mr. O. Z. Howard, of the Engineering Department, Naval Academy, for his painstaking care in the drawings and assistance in revision.

JOHN K. BARTON,
Commander, U. S. Navy.

U. S. NAVAL ACADEMY, *Annapolis, Md.*, June, 1904.

PREFACE TO THIRD EDITION

In revising this work it was decided to omit turbines, since that subject is dealt with at the Naval Academy in a text book devoted to turbines solely; thus limiting this edition, in so far as concerns main engines, to reciprocating machinery only.

Sincere thanks are due to Captain C. W. Dyson, Commander U. T. Holmes, and others in the Bureau of Steam Engineering for assistance rendered; to manufacturers of auxiliaries for improved details; to Chief Machinist W. T. Robinson and Machinist Max Vogt for assistance in revision of Chapters 9, 14, 17 and 18; and to Mr. F. H. Rittenour for preparation of drawings.

H. O. STICKNEY,
Commander, U. S. Navy.

U. S. NAVAL ACADEMY, *Annapolis, Md.*, June, 1913.

In the preparation of this book, the following have been consulted:

Journal of the American Society of Naval Engineers.

Machinery specifications for U. S. naval vessels.

Annual Reports of the Chief of Bureau of Steam Engineering,
U. S. Navy.

A Manual of Marine Engineering, by A. E. Seaton.

The Marine Steam Engine, by R. Sennett and H. J. Oram.

Steam Engine, Theory and Practice, by Professor W. Ripper.

Manual of Naval Architecture, by W. H. White.

Resistance of Ships and Screw Propulsion, by D. W. Taylor.

The Steam Engine, by J. A. Ewing.

Practical Marine Engineering, by W. F. Durand.

The Steam Engine, by G. C. V. Holmes.

A Treatise on the Steam Engine, by Arthur Riggs.

Marine Propellers, by S. W. Barnaby.

Manual of the Steam Engine Indicator, by C. H. Peabody.

Steam and the Marine Steam Engine, by J. Yeo.

Engine Room Practice, by J. G. Liversidge, R. N.

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CHAPTER I.

WORK—ENERGY—POWER.

Work consists in overcoming resistance through space; the result of the exertion of force through a certain distance. In order that work may be done there must be, not only a force, but also a motion.

Energy is defined as the power or capacity of bodies for doing work, and a body has more energy the more work it can do without regarding time. A force acting through space can be said to exert energy, but when this is applied and a resistance overcome through a certain distance, then work is done.

The unit of work is the *foot-pound*, or the work done by a force of one pound acting through the space of one foot; thus 10 pounds raised through a distance of 5 feet, or a pressure of 5 pounds exerted through 10 feet, produces in either case 50 foot-pounds of work. The *foot-ton* or *inch-ton* is sometimes employed for convenience in special cases and is the work of lifting one ton through one foot or one inch. The unit of work has no reference to the time taken, for the same amount of work is done in overcoming the resistance whether it can be done in one minute or one hour.

From the definition of the unit of work it follows that the work done by a body equals the product of the mean force of resistance in pounds, and the distance moved in feet. Hence, it may be represented by the *area* of a closed *figure*, which is also the product of two quantities, the mean altitude representing the mean resistance force, and the length, the space or distance moved through. This is known as a *work diagram* and is the method practically applied in measuring the work done by engines, and for which the indicator is used. (See Chapter X, The Indicator and the Indicator Diagram.)

Power.—The power of an engine involves the element of time, and is measured by the rate at which it can do work, and is the amount of work done in a unit of time.

In engineering calculations, the unit of power adopted is the *horse power* which represents the performance of 33,000 foot-pounds of work per minute or 550 foot-pounds of work per second. Since the heat unit is equal to 778 units of work, one horse power is equal to $33,000 \div 778 = 42.42$ B. T. U. per minute.

$$\text{Horse Power} = \frac{\text{Work done}}{33,000 \times \text{time in minutes.}}$$

Energy is defined as the capacity for doing work and must not be confused with *Power*. A body is said to contain more energy the more work it can do regardless of time. One ton of coal on being burnt in the furnace of a boiler can produce steam for developing, say 700 horse power in one hour, whereas in another smaller boiler it can produce 50 horse power for 14 hours.

Energy, whether heat energy or any other, cannot be destroyed. It can take different forms, but according to the principle of the *Conservation of Energy*, the total of the energy remains the same; the heat carried to the engine in the steam is transformed into work, some passes to waste in various ways, but the sum of the heat usefully employed plus the heat carried to waste always equals precisely the heat evolved from the source.

Efficiency.—Of the entire energy received by an engine, some is utilized in the production of useful work and the rest is lost or wasted, in overcoming friction for instance, and in various other ways. The ratio of the useful work done to that of the energy supplied is known as efficiency; thus in the case of a steam plant, it is the ratio of the heat changed into work and the entire heat supplied.

$$\text{Efficiency} = \frac{\text{Useful work done}}{\text{Total energy received.}}$$

In the motive power of a ship the final work done is the effect in driving the ship. Compared with the heat energy contained in the fuel consumed in this operation, the work done will be found to be but a small fraction, and is due to the various losses from the boiler through the mechanism to the propeller. These losses are made up of (1) losses in the boiler, (2) of the heat carried to waste in the steam itself after leaving the boiler, (3) losses in the

mechanism, and (4) propeller losses. The total efficiency of the propelling apparatus is the product of the factors representing the efficiencies of these four elements. Any gain in the efficiency of one part results in a corresponding gain in the general economy of the whole machine.

In the text-book on Naval Boilers, it has been shown that the principal losses in the boiler are due to incomplete combustion, condensation, radiation and high temperature of smoke-pipe gases, so that only 60 to 70 per cent of the total energy of the fuel is converted into useful work. After steam leaves the boiler further losses are experienced as the steam traverses the cylinders, due to radiation, incomplete expansion, the action of the cylinder walls, and principally the heat carried to waste in the condenser. The ratio of the effective work performed on the piston to the total heat contained in the steam does not amount to more than from 10 to 20 per cent.

All the work made available by the steam driving the piston does not reach the propeller, owing to losses in the mechanism, such as frictions, etc., the energy wasted in this operation amounting to about 20 per cent, leaving for the efficiency of the mechanism about 80 per cent. Of the work now left to drive the propeller not more than 50 or 60 per cent is utilized in overcoming the resistance to the vessel and driving her ahead, owing to the losses in frictional and edgewise resistance of the water on the blades, augment of resistance and the slip.

From the efficiencies given for the several stages from the boiler to the propeller, it will be seen that the maximum efficiency of the entire propelling machinery under the present conditions does not exceed 5 or 6 per cent.

In the succeeding chapters will be taken up the causes of the losses in the steam, mechanism and propeller; to what extent they are unavoidable; and the means taken to reduce them to a minimum.

Before proceeding to discuss the action of the steam in an engine, an elementary description of its parts is essential in order to understand the mechanism used in the conversion of the heat energy of the steam into the work of driving the engine.

In Fig. 1, steam from the boiler is conveyed through the steam pipes to the engine stop valve *T*. From *T* a certain quantity goes through the *main valve v* into the *cylinder 1*, and there exerts energy against a *piston P*. By suitable mechanism, called the *valve gear*, *v* is actuated to admit steam to one end of the cylinder and to allow it to escape or *exhaust* from the other end, the piston *P* being moved by the steam during this time from one end of the cylinder to the other. This *reciprocating motion* of *P* is converted outside of the cylinder by suitable mechanism into a *turning motion* of the shaft *a*, to the end of which is secured the

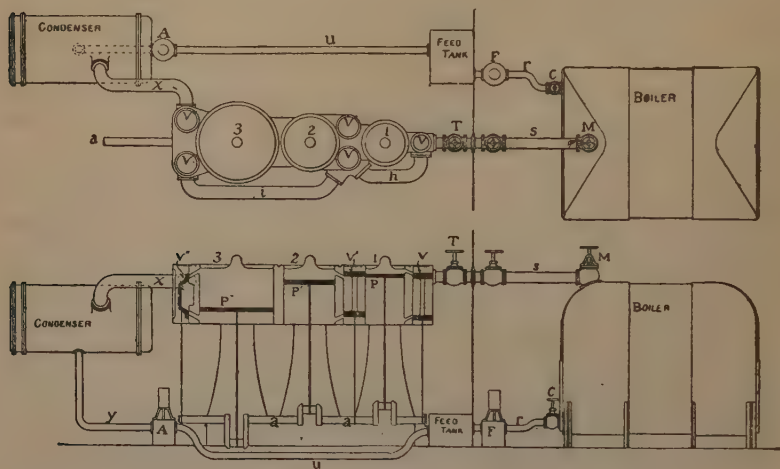


FIG. 1.

propeller. The water in which the latter turns and engine friction form the chief resistance which the steam acting on *P* has to overcome.

After acting on *P* in one direction throughout its *stroke*, or distance that *P* can travel in the cylinder, the original quantity of steam is exhausted through one or more additional cylinders into the *condenser*, or it may be exhausted from cylinder 1 direct into the condenser, or, more directly, into the atmosphere. In this last case, the engine is called *non-condensing*.

If only one cylinder and a condenser are used, they form a

simple condensing engine. When more than one is used, each has its valve, piston and other mechanism, the steam going through the same action in each cylinder successively. The cylinders, valves and pistons increase in size for each additional cylinder, and a condenser is always used, the reasons for which will be shown later. The pipes, *h*, *i*, and other spaces connecting the exhaust side of one cylinder to the steam admission of the next, are called *receivers*.

Engines in which the steam passes successively through two, three or four cylinders are called *double*, *triple* and *quadruple* expansion engines, the first one being commonly called a *compound* engine. The cylinder which receives the boiler steam is called the *high-pressure* cylinder, and that which exhausts it into the condenser, the *low-pressure* cylinder. The cylinders between these two are called *intermediate-pressure* cylinders. The various cylinders are designated by the explanatory letters, H. P., I. P., and L. P. For a quadruple expansion engine, the I. P. cylinder next to the H. P. is designated the 1st I. P., and the next one as the 2d I. P.

In the condenser, the steam is converted into water, which with any air is removed by means of the *air pump A* and returned to the boiler through a *feed tank* and by the *feed pump F*.

CHAPTER II.

THE ACTION OF THE STEAM.

From the definition, the efficiency of the steam is simply the ratio of the energy exerted by the steam to the heat required to produce it, and since the difference between the amounts of heat utilized to obtain the same weight of steam at different pressures is so slight, the question of economy is only that of obtaining the greatest amount of work from a given weight of steam.

From the formula for total heat of steam, it will be seen that the amount of heat required to convert one pound of water from a temperature of 100° into steam of 60 pounds pressure, is about 1115 thermal units, and to convert the same quantity of water, at the same temperature, into steam at 200 pounds pressure, is 1140 thermal units, only $2\frac{1}{2}$ per cent greater heat required for three times the pressure. This fact is utilized in modern engines by the employment of high pressures and expansion, with a consequent increase of work from the same weight of steam.

A well-known formula and theorem of thermodynamics, showing the maximum efficiency of steam, is represented by

$$E = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{t_1 + 461}$$

where T_1 = initial absolute temperature; t_1 = initial temp. F.; T_2 = final absolute temperature; t_2 = final temperature F.

That is, the greatest possible efficiency of any heat engine depends only upon the highest and lowest temperatures between which it is worked, and is independent of the nature of the working substance. The higher the temperature and pressure of the working fluid supplied and the lower that at which it is rejected, the higher will be the efficiency of the engine.

Taking the case of a modern engine working with saturated steam of an initial pressure of 250 pounds per gage, at the engine, approximately 265 pounds absolute, and exhausting into a con-

denser in which there is an absolute pressure of 3 pounds per square inch, approximately equal to a vacuum of 24 inches, the corresponding initial and final temperatures will be 401° F. and 141° F., assuming that there is no difference between the back pressure or pressure on the exhaust side of the piston and the pressure in the condenser.

$$\text{Then, Maximum Efficiency} = \frac{401 - 141}{862} = .301.$$

That is, even if the total heat of the saturated steam admitted at a temperature of 401° F. could be so utilized in work in the engine, that, when the steam leaves the engine, it would carry away only the total heat due to the temperature of 141° F., the greatest possible amount of heat utilized would be only about one-third of that supplied. It would be, therefore, impossible to reduce the loss, even in an ideally perfect engine, below 70 per cent for the temperature range given above. But in the actual engine the ideal conditions are not reached, due to losses which will be taken up later, so that instead of a loss of 70 per cent, practically it amounts to 75 or 85 per cent, making the efficiency of a good steam engine only from 15 to 25 per cent.

Methods of Increasing Efficiency.—To increase the range of temperature and hence the efficiency, the value of t_1 has been successively raised, as boilers could be designed to withstand the increased pressures, from the former low pressures of 25 and 30 pounds up to 250 and 300 pounds of the present day. In looking for a reduction of t_2 , it is impossible with a non-condensing engine, or one exhausting into the atmosphere, to obtain a much smaller back pressure than 18 or 19 pounds absolute, or 3 or 4 pounds above the pressure of the atmosphere, owing to the friction in pipes and passages and the speed of the engine. The lower value of t_2 in the non-condensing engine is consequently limited to 220° F.

In the condensing engine, however, in which the exhaust steam is discharged into a condenser, and condensed by cooling water, the temperature and pressure are both lowered. The back pressure in this case falls to about 3 pounds absolute and a corresponding temperature of about 141° F. The difference between the back

pressure and that of the atmosphere is called a vacuum and is measured in inches of mercury.

As an example of this saving by the use of a condenser, let the initial steam have a pressure of 250 pounds absolute with a corresponding temperature of 401° F.

$$\text{For non-condensing } E = \frac{401 - 220}{401 + 461} = .209.$$

$$\text{For condensing } E = \frac{401 - 141}{401 + 461} = .301,$$

or the efficiency of the steam has been increased 45 per cent by the installation of a condenser.

The above result can be shown graphically by means of the indicator diagram. This diagram is obtained from an instrument attached to the cylinders of an engine and shows the pressures in the cylinders at varying points of the stroke, and many other facts of importance regarding the action of the steam within the cylinder. (See Chapter X.)

The theoretical form of an indicator diagram taken from a condensing engine, in which a gas such as superheated steam is used, the back pressure assumed constant throughout the stroke and the admission and exhaust of steam instantaneous, is shown in Fig. 2.

Ordinates represent the steam pressures while the abscissa represents the volume. Since the volume of a cylinder is the product of the stroke and area of piston, the line of volume may also be used for the stroke.

Let OV represent the *stroke* of the engine. The vertical *admission line* on the left shows the rise of pressure and OP represents the absolute initial pressure on the piston. As the piston moves with this pressure constantly upon it, the horizontal *steam line* PA is traced; at A the admission of steam is cut off by the valve closing the steam port. Expansion commences at A , the piston being driven forward by the expansive action of the steam, and the pressure falling as the volume increases. This period is represented by the *expansion line* AB . Had the steam followed with full steam pressure to the end of the stroke, the admission line would have been carried to A' . At A' (for a non-expansive) or

B (for an expansive engine) the exhaust valve opens to the atmosphere or condenser as the case may be and the pressure falls to the final one as shown VK or VE . The line BK or BE is called the exhaust line. The piston now begins to return, steam of initial pressure being admitted to the other side, and the steam escapes against the back pressure VK or VE until the end of the return stroke at O . For a condensing engine, DE is called the back pressure line. The exhaust now closes and live steam is again admitted and the operation repeated for each stroke.

By closing the indicator cock to the cylinder and opening to the atmosphere, the atmospheric line GH is drawn.

In engine calculations, the pressures used are absolute steam pressures or pressures measured from the zero line or line of no

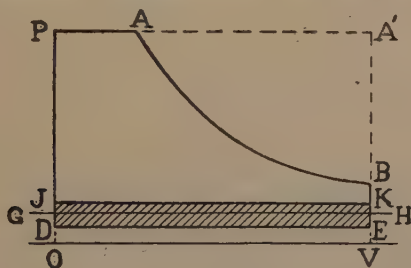


FIG. 2.

pressure, and should be distinguished from *pressures by gage*, which latter are simply steam pressures above that of the atmosphere, and are those pressures indicated by the readings on the ordinary pressure gages.

Since the indicator diagram is merely a diagram of work, its length shows the distance moved through by the piston, while the effective force is the excess of the mean forward pressure against the piston on one side, during admission and expansion, above the mean back pressure on the other side, as the exhaust steam passes to the condenser. Hence the area of the entire diagram $OPABV$, minus the area of back pressure $ODEV$, represents to scale the effective work done on one side of the piston. A similar diagram taken from the other end of the cylinder shows the work done on the other side during the double stroke.

The area of this diagram, and consequently the work done with a given *weight of steam*, may be increased by decreasing the amount of back pressure, *i. e.*, lowering the back pressure line, also by raising the steam and expansion lines. The former can be done by the use of a condenser and the latter by increasing the initial pressure and using an earlier cut-off.

The comparative work of the two engines mentioned on page 22 can be shown by the diagram, Fig. 2, since the initial pressure and the cut-off are the same, the only difference in the cards from each engine would be the position of the back pressure line. In the non-condensing engine this line would be at *JK*, and for the condensing engine at *DE*, giving an increase in the area of the condensing engine diagram, and, consequently, an increase in the work done represented by the area *JKED*.

Definitions.—*Absolute Temperature.*—Temperatures, estimated not from the ordinary zero, but from a point 461° below the zero of Fahrenheit's scale, are known as *absolute temperatures*, and the point, — 461° F. is called the *absolute zero of temperature*.

Absolute Pressure is measured from the zero pressure of the atmosphere or 14.7 pounds per square inch below the pressure of the atmosphere when the mercury barometer stands at 30 inches.

Specific Volume.—The specific volume of steam is the number of cubic feet contained in one pound of steam at any given pressure.

Relative Volume.—The relative volume is the ratio which the volume of steam produced bears to the volume from which it was generated.

Ratio of Expansion is the ratio which the volume of a given quantity of steam after expansion bears to the volume it occupied before expansion.

Saturated Steam is steam at the greatest possible density for its pressure. Steam in this condition is just at the point of condensation and any increase of pressure or decrease of temperature will cause some of it to be condensed. The temperature of saturated steam is the temperature of the water with which it is in contact. For each pressure of saturated steam there is a fixed temperature. See "Saturated Steam Table" in Appendix.

Many formulæ have been produced to represent the results of Regnault and other experimenters on the relation between the pressure and the temperature of saturated steam. They are all of a complicated nature; the simplest is probably the following:

$$\log p = 5 \times \frac{t - 212}{t + 367} + \log 14.7,$$

where p is the absolute pressure in pounds per square inch and t is the temperature in degrees Fahrenheit.

For ready numerical calculations, a steam table, such as given in the Appendix, will be found more useful and convenient, but in the absence of such a table the above formula can be used for all practical purposes.

The relation between the pressure and volume of saturated steam may be approximately expressed by means of the formula:

$$pv^{1\frac{1}{2}} = 475, \text{ or more accurately, } pv^{1.0646} = 479,$$

where p is the absolute pressure in pounds per square inch and v is the volume in cubic feet.

Superheated Steam.—Steam free from the water from which it was generated and supplied with additional heat becomes *superheated* and approaches the condition of a perfect gas, while the temperature and density no longer correspond with the pressure as during evaporation. The temperature of saturated steam in the presence of water cannot be raised without raising its pressure, but steam may be superheated without an increase of pressure, if it is free from water and is allowed to expand as the heat is added.

Wet Steam.—Ordinary steam carrying with it a greater or less proportion of water, either in suspension or otherwise mixed with it, is known as *wet or moist steam*. Wet steam will become drier or saturated steam become superheated if it falls in pressure without a corresponding loss of heat, as for instance, in the free expansion of the exhaust steam to the receivers of a multiple expansion engine.

Expansive Action of Steam.—From the expansive property which steam and other gases possess, a still further economy may be obtained by the performance of increased work from the same weight of steam. By taking advantage of this property and cut-

pounds, the exhaust opens and allows the pressure to fall to L . The work done is now the sum of the admission work and the work of expansion, shown by the rectangle OA plus the figure V_1AEV_2 minus the work of back pressure shown by the rectangle OL .

If, instead of cutting off at $\frac{2}{10}$ of the stroke, the steam had been carried full pressure up to $\frac{5}{10}$, or at M , the expansion curve would now be MN and the terminal pressure 125 pounds. The effective work in this case is shown by $OM + KMN V_2 - OL$. If steam had not been cut off, but allowed to follow full stroke, the diagram would take the form of the rectangle OA' , and the effective work would be indicated by the area $OA' - OL$.

In the above cases when steam is allowed to expand it is apparent that the work represented by the area between the expansion and vacuum lines represents a gain in work, since after cut-off no more steam is used for the stroke. This area is increased with the same weight of steam, the higher the initial pressure and the greater the expansion, as will be shown later.

To obtain a numerical value for the gain in the above cases the area of the expansion part of the diagram must be ascertained.

On the assumption that the steam used approaches a perfect gas and that $pv = \text{constant}$, the expansion curve is a rectangular hyperbola and the exact value of the expansion part of the diagram may be obtained from a table of hyperbolic logarithms, for when the curve is hyperbolic, the hyperbolic logarithm expresses the relation between the area during expansion, and the area during admission. This curve does not apply exactly to steam, but approximately agrees with that found in practice. It is, however, simpler than the others and sufficiently accurate for the purpose. In the absence of a table of hyperbolic logarithms, the $\log_e$ of a number may be found by multiplying the common logarithm of the number by 2.3026.

Let V_1 equal the initial volume of steam before cut-off, and V_2 the final volume, and r the ratio of expansion, or $r = \frac{V_2}{V_1}$. The point of cut-off is the reciprocal of r .

Let p_1 , p_2 and p_3 equal the initial, terminal and back pressure, respectively. In the case when the steam is cut off at $\frac{2}{10}$:

$$\text{Admission work, } OA = p_1 v_1 \quad (1)$$

$$\text{Expansion work, } v_1 A E V_2 = p_1 v_1 \log_e r \quad (2)$$

$$\text{Total forward work, } (1) + (2) = p_1 v_1 (1 + \log_e r) \quad (3)$$

The effective work equals the total forward work minus the work in overcoming the back pressure.

$$\text{Effective work} = p_1 v_1 (1 + \log_e r) - p_3 v_2.$$

Substituting values of p_1 and v_1 when cutting off at $\frac{2}{10}$, $\frac{5}{10}$ and full stroke and neglecting the uniform back pressure in each case,

$$r = 5; \text{ Work done} = 250 \times .2(1 + 1.609) = 130.45,$$

$$r = 2; \text{ Work done} = 250 \times .5(1 + .693) = 211.62,$$

$$r = 1; \text{ Work done} = 250 \times 1(1 + 0) = 250.$$

That is, with but $\frac{1}{5}$ of the quantity of the steam used the work done is over one-half that done by the whole weight of steam when not used expansively or considering *equal weights of steam* used, the gain by expansion, when cutting off at .2, would be 160.9 per cent, and for a cut-off at one-half stroke 69.3 per cent.

Gain by Expansion with Increased Pressure.—In the previous questions a constant initial pressure was maintained and the amount of steam used varied. To show the gain by expansion when the *same weight* of steam is used, but with increased pressures, refer to Fig. 4.

Let one pound of steam of 50 pounds absolute pressure be admitted to a cylinder for the entire stroke OV_4 , then the work done = OD . Next, let the steam be admitted at 100 pounds and cut off at one-half stroke, the work now done by the steam is represented by $OGCDV_4$. Assuming that the steam obeys Boyle's law, the weight of the steam used is the same in both cases. Similarly, if the steam is admitted at 200 and 250 pounds and volumes V_2 and V_1 respectively, the work is now represented by $OHBDV_4$ and $OPADV_4$, the effective work in each case being reduced by the uniform back pressure area OK .

The gain by expansion in each case is shown by the whole figure, minus the area representing the work done by the same weight of steam used without expansion, that is the area OD .

Calling the work represented by OA unity, and neglecting the back pressure, the expansion area, or the gain in work during expansion, will be equal to the hyperbolic logarithm of the ratio of expansion in each case.

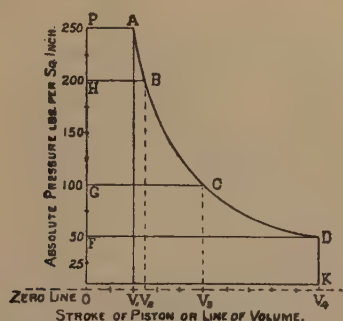


FIG. 4.

$p_1 = 250$; $r = 5$	gain during expansion	1.609,
$p_1 = 200$; $r = 4$	gain during expansion	1.386,
$p_1 = 100$; $r = 2$	gain during expansion	.693,
$p_1 = 50$; $r = 1$	gain during expansion	.000.

With higher pressures and increased rates of expansion, it will be seen from the foregoing that the gain in economy is increased, though the further gain becomes less and less for a given increase in expansion; for instance, in increasing p_1 from 50 to 100 pounds the gain is .693, while for the same amount of increase from 200 to 250, the gain is only .223.

Limit of Expansion.—As long as the forward pressure on the piston exceeds the back pressure, energy would continue to be exerted and in order to obtain economy from the steam, the expansion should be such as to bring the expansion line down to the back pressure line. In practice it is not possible to carry the expansion so far, owing to further losses, such as cylinder condensation and friction of the mechanism. There is a practical limit to the economy indicated by the diagram, and expansion may be carried so far as to increase the amount of steam used.

In explaining the action of steam it has been assumed that it acts as a perfect gas and does work on a piston, while expanding in a non-conducting cylinder, and during this expansion the tem-

perature remains constant. To maintain this constant temperature, heat must be added to the steam from an outside source. These conditions do not obtain in practice; the steam in a boiler is not a perfect gas, nor is the cylinder a non-conducting, non-radiating cylinder, so that any loss due to work on the piston, radiation or conduction, occasions the liquefaction of some of the steam, with the consequent loss of volume and pressure.

In discussing the laws of the expansion of steam, three cases are usually considered, varying with the character of the expansion curve after cut-off, the nature of this curve depending upon the conditions as to the gain or loss of heat during expansion.

Case I—The Hyperbolic Curve.—When damp or moist steam enters a steam-jacketed cylinder, a large quantity of heat passes from the jacket to the cylinder steam. If this amount of heat is not only equivalent to the work performed, but sufficient to convert a portion of the damp steam into dry saturated steam during expansion, the relation between the pressure and volume is approximately expressed by Boyle's Law, viz.:

$$pv = \text{constant.}$$

This curve is shown at *I*, Fig. 5, and approximately agrees with that obtained in practice; from this fact and from the ease with which calculations can be made, the hyperbolic curve is the one usually applied for approximate work on engine diagrams.

$$p_m = p_1 \left(\frac{1 + \log_e r}{r} \right)$$

$$p_2 = \frac{p_1}{r} \text{ and } p_e = p_m - p_3,$$

when p_1 = initial absolute pressure, p_2 terminal pressure absolute, p_3 absolute back pressure, p_m = mean pressure and p_e = mean effective pressure.

Case II—The Saturated Curve.—When dry saturated steam expands doing work, but receiving heat during expansion from some external source, such as a steam jacket, sufficient to prevent liquefaction, the curve falls below the hyperbolic curve, since the temperature does not remain constant, but falls to the temperature corresponding to the lowered pressure. The pressure at all

points of the stroke is that due to the temperature and volume of saturated steam. This curve of expansion is usually known as the *saturated curve*, and the formula given by Prof. Rankine for the pressure is

$$pv^{\frac{1}{2}} = \text{constant}$$

$$p_m = p_1 \times \left(\frac{17 - 16r - \frac{1}{2}}{r} \right)$$

$$p_2 = p_1 r^{-\frac{1}{2}}$$

Case III—The Adiabatic Curve.—When saturated steam expands in a non-conducting, non-radiating cylinder and during ex-

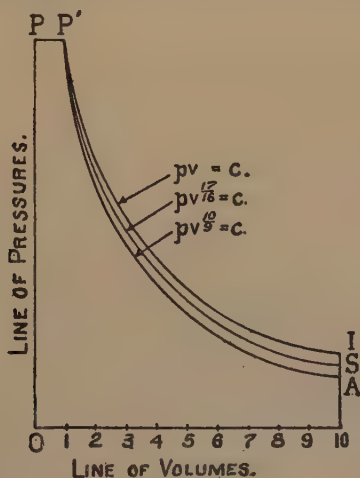


FIG. 5.

pansion performs work as before, without receiving or parting with heat during expansion (except in the performance of work) the pressure falls both from increase of volume, and liquefaction due to the performance of work, and much below that shown by the isothermal and saturation curves.

Rankine's approximate formula for this curve is $pv^{\frac{5}{9}} = \text{constant}$.

$$p_m = p_1 \times \frac{10 - 9r - \frac{1}{9}}{r}$$

$$p_2 = p_1 r^{-\frac{5}{9}}$$

These curves of expansion are shown in Fig. 5; PP' is the initial volume, $O-10$ the final volume, $P'I$ is the isothermal or

hyperbolic, $P'S$ the saturated, and $P'A$ the adiabatic curve of expansion. As stated above, Case I is generally used in practical diagrams. Case II is frequently used in the analysis of indicator diagrams; the formula involves a large amount of calculations, but by the use of Steam Tables this work can be greatly abridged. Case III is used in certain theoretical investigations and work of comparison.*

* To find the mean pressure (p_m) when a gas expands according to any law.

Let p_1 = initial pressure, p_2 = terminal pressure, p_m = mean for'd pressure, v_1 = initial volume, v_2 = final volume, r = ratio of expansion = $\frac{v_2}{v_1}$, or $v_2 = rv_1$, p_m = total area of card (Fig. 5-A) divided by the length or $p_m = \frac{A}{v_2} = \frac{A}{rv_1}$.

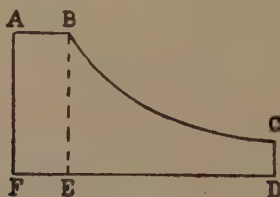


Fig. 5-A.

Assume the equation of the expansion curve to be $pv^n = C$, then the area

$$BCDE = \int_{v_1}^{v_2} p dv \quad (1)$$

But $pv^n = p_1v_1^n = p_2v_2^n$, etc., hence $p = \frac{p_1v_1^n}{v^n}$

$$\text{Substituting in (1) area } BCDE = \int_{v_1}^{v_2} p dv = \int_{v_1}^{v_2} p_1v_1^n \left(\frac{dv}{v^n} \right) \quad (2)$$

Adding to this the area of the rectangular portion of the diagram $ABEF$, we obtain for the entire area of the whole diagram $ABCDEF = p_1v_1 + p_1v_1 \int_{v_1}^{v_2} \frac{dv}{v^n} \quad (3)$

Integrating this latter expression between the limits v_1 and v_2 or v_1r , we obtain a value for the work area.

In the special case when the expansion curve is hyperbolic and $n=1$ (Case I) integrating and solving we have, work area = $p_1v_1 + p_1v_1 (\log_e v_1r - \log_e v_1) = p_1v_1 (1 + \log_e r)$,

$$\text{then since } p_m = \frac{\text{Work Area}}{rv_1}, p_m = \frac{p_1v_1(1 + \log_e r)}{rv_1} = p_1 \times \frac{1 + \log_e r}{r}$$

similarly for the saturation and adiabatic curves after performing the integration demanded in equation (3), the following expressions for p_m are obtained:

$$p_m = \frac{p_1}{r} \left(1 + \frac{r^{1-n} - 1}{1-n} \right), \text{ or } p_m = p_1 \times \frac{r^{1-n} - n}{r(1-n)}.$$

Method of Drawing the Hyperbolic Curve.—In Fig. 6 draw the absolute zero line OF and at right angles to it, the line of absolute pressures OP . The line OF is drawn through the zero of pressure, a distance below the atmospheric line equal to the atmospheric pressure. From the point O lay off OA equal to the volume of the cylinder up to the point of cut-off including clearance and OP equal to the initial absolute pressure. Complete the rectangle $OPaA$ and prolong the line Pa parallel to the zero line. To find the pressures at the points B, C, D, E , corresponding to the cylinder volumes OB, OC, OD, OE , draw a perpendicular at B cutting the line Pa produced at b . Join Ob , cutting the line Aa

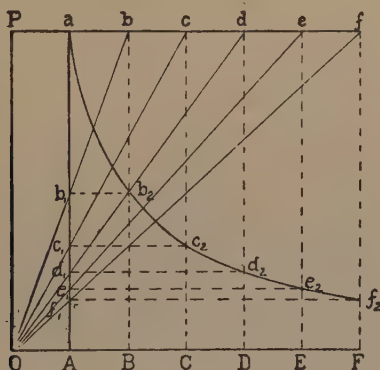


FIG. 6.

at the point b_1 . Through the point b_1 draw a line parallel to the zero line, cutting the vertical Bb at b_2 and giving one point in the hyperbolic curve. Similarly the points c_2, d_2 and e_2 may be obtained and the completed curve drawn through the points.

The saturation curve can be drawn from the table of properties of saturated steam as given in Table XV, ordinates to represent the various pressures and the corresponding volumes taken from the table to represent the abscissæ. The adiabatic curve can be laid down by substituting values in the formula given and solving for different points of the curve, or by construction, from the knowledge that the horizontal distances of the adiabatic curve from the saturation curve are equal to that of the latter from the hyperbolic.

Liquefaction, Initial Cylinder Condensation.—Excepting the loss of heat carried to the condenser by the exhaust steam, initial condensation of steam in the cylinders is the most serious loss connected with the use of steam as a working medium. Throughout the stroke, during admission, expansion and exhaust, there is a complex interchange of heat between the cylinder and the steam, greatly increasing the actual steam consumption and consequently reducing the efficiency. The action of the steam in general is as follows:

The steam from the boiler is admitted to the cylinder at the beginning of the stroke and coming in contact with the metallic surfaces of the cylinder and piston, which have been chilled by contact with the cool steam of expansion and exhaust of the previous stroke, about 140° in temperature, condensation immediately commences and continues as the piston advances, until a point is reached when the temperature of the cylinder walls, piston and cylinder head approach that of the entering steam. By this time the surfaces are covered with a film of moisture, with spray and moisture intermixed with the steam in the cylinder, which owing to its conductivity greatly facilitates the exchange of heat between the steam and the metal of the cylinder walls.

Since communication with the boiler is open, the place of the steam liquefied is supplied by fresh steam from the boiler, so that at the point of cut-off a much greater supply of steam has been taken from the boiler than corresponds to the volume of the admission space. After cut-off additional condensation takes place due to the work done during expansion. As the expansion proceeds the steam pressure falls, the deposited moisture and water begins to re-evaporate as soon as the temperature of the steam falls below that of the deposited water and cylinder walls, and the weight of the steam present in the cylinder as steam gradually increases towards the end of the stroke. This heat transferred to the steam by re-evaporation *during expansion* and before exhaust is not a loss, as work is done on the piston by this re-evaporated steam before exhaust opening. In extreme cases when the cut-off is very short and the initial condensation is very great, the cylinder is liable to become choked with water, the *cylinder relief* valves

must then be opened to prevent cracking the cylinder heads, and a further waste ensues.

During the exhaust period, the pressure has dropped to that of the condenser and the water boils at a temperature of about 140° , greatly accelerating the re-evaporation, until the cylinder walls have reached the temperature due to the back pressure. This heat transferred to the steam *during exhaust* goes direct to the condenser without doing any useful work, increases the back pressure to an appreciable extent and causes a serious loss in the efficiency of the steam.

In general, cylinder condensation and consequent loss from liquefaction will be increased by an increase in the range of temperature between admission and exhaust; by increased surface exposed to a given weight of steam, or by any increase of time for the interchange of heat to take place. Consequently, if the ratio of expansion is increased, cylinder condensation is increased; high-speed engines should show more economy, other conditions being the same, than low-speed engines. The size of the cylinders also influences the condensation, the loss being proportionately greater in small engines, since the area of the *surface* of the cylinder wall is, relatively to volume, greater for a smaller diameter.

Methods of Reducing Liquefaction Losses.—The various methods adopted for reducing the loss from liquefaction and consequently increasing the efficiency of the steam in the cylinder are as follows:

1. Steam-jacketing the cylinders.
2. Superheating the steam before admitting to cylinder.
3. Reheaters in the receivers.
4. Compression.
5. By the use of stage expansion in different cylinders as in double, triple or quadruple expansion engines.
6. Increasing the speed of rotation.

Steam Jacketing.—This consists in heating the cylinder by passing hot steam from the boiler into an annular space around the barrel, and frequently into the cylinder covers, which are made hollow. By keeping this space filled with steam at a temperature

equal to or greater than the initial steam in the cylinder, the alternate heating and cooling of the metallic surfaces, just shown to take place, is greatly reduced and the steam enters a cylinder nearly as hot as itself. The initial condensation, and condensation during expansion in the performance of work is thus largely reduced or prevented, and the transfer of heat as heat from the jacket steam is often sufficient to keep the steam in a saturated condition throughout the stroke.

This heat transmitted from the jacket through the cylinder walls is accompanied by a corresponding loss due to condensation in the jacket, but here it produces no bad effects, the condensed steam being led off by drain pipes to traps and thence to the feed tanks and direct to the boiler.

There is a partial loss in the jacket from the fact that steam used there must be taken from the boiler, but this is largely counterbalanced by reducing the more serious loss of heat which occurs through the *alternate heating and cooling* of the cylinder walls. The net result, however, is in favor of jacketing, for the jacket drains carry away *water* at its sensible temperature, while the steam condensed in the cylinder passes away as *steam*, carrying to the condenser the *latent heat* which had been first given to the cylinder metal on admission, and then taken from it on re-evaporation. This has been proved by numerous engine tests, the saving having been always in excess of the amount of heat expended in the jacket, the net saving amounting to as much as 10 and 20 per cent.

The success of jacketing depends largely upon the design of the jacket and the care with which it is used. There should be no pockets to fill up with condensed steam, the lead of pipes to drain off water must be good, and circulation of steam through the jackets must be free.

The economy of the jackets for various rates of expansion and for various engines has been the object of many experiments, from the numerous and valuable ones on ship engines made in and for our navy by Chief Engineer Isherwood and Mr. Emery to those by the Research Committee on Steam Jackets for the British Institution of Mechanical Engineers. Besides these, valuable tests

have been made by individual experimenters. From the results of these experiments the following summary may be made:

1. The greater the speed of rotation, the less will be the value of the jacket. This could be expected, for the amount of heat transmitted is proportional to the time; as the number of strokes increase per minute, the transmission per stroke decreases, until at an infinitely great speed the transmission is zero.

2. The value of the jacket is less as the size of the engine increases, for the *volume* of the cylinder increases as the *square* of the diameter and the *area* of the surface increases only as the diameter.

3. For the same ratio of expansion the value and necessity of the jacket decreases as the number of cylinders in which this expansion takes place increases.

4. The jacket should be filled with steam of a much higher temperature than that admitted to the cylinder. There must be quite a difference of temperature to induce a flow of heat, but the practical difficulties of lubrication and cylinder scoring limit this difference.

Steam should not be passed through the jacket on its way to the cylinder, as it then would be partially condensed before admission to the cylinder and its efficiency impaired by the presence of water.

There is one great advantage of the jacket, which although it does not affect the economy or efficiency of the steam, may be mentioned. By its use the engines can be warmed before starting, and kept warm and ready for use, without admitting steam into the cylinders.

The present practice in our navy, and extending back some years, is to jacket the liners of all cylinders, and, in addition, both heads of the larger I. P. and L. P. cylinders. It will be found, after running for some time, if the pistons are tight, that the surface of the liner is clean and bright, while the surfaces of the pistons, heads and passages are covered with a good non-conducting deposit. The latter will, therefore, interfere materially with any useful effect of the jacketed heads during running.

Jackets to be efficient must have positive circulation of steam. To effect this, in the Oklahoma the jacket steam and drain pipes are unusually large, and steam after passing successively through the jackets goes from the L. P. to a separator, thence to a circulating pump engine. Thus a steady flow of steam is induced, water is prevented from gathering in the jackets and greater efficiency will result.

Reheaters.—These consist of coils or tubes placed in the receivers, through which steam circulates. The effect is similar to that of the jacket, and what has been said about the jacket applies to reheaters. Reheaters are not now used in marine engines except when reciprocating engines are combined with turbines they may be used to dry the exhaust from the L. P. engine before it goes to the turbine.

Superheating the steam before its admission lessens the quantity of latent heat required to furnish the heat taken up by the cylinder walls, for if the steam is sufficiently superheated it can part with the whole of its superheat without undergoing any liquefaction. Under these conditions superheating adds greatly to the economy by reducing liquefaction in the cylinder, but practical considerations have so far prevented a greater degree of superheating than is necessary to provide dry steam as far as the engine.

When the steam was very much superheated it was found to have a very injurious effect; the internal lubricants became burnt and the packing destroyed, but in the modern engine where little or no internal lubrication is used and metallic packing employed, it is not so much the high temperatures that cause injury as it is the extreme dryness of superheated steam and the absence of the lubricating effects of a slight moisture, causing the piston and valve to work dry, cutting cylinder and valve faces and inducing leakage. Superheating at the boilers for about 100° F. has been found beneficial and will generally be sufficient to dry the steam or make up for the condensation in the steam pipes between the boilers and the engine.

Low-pressure steam was usually superheated by causing it to pass through or among tubes placed in the uptakes of the boilers, the necessary heat being obtained from the waste gases on their way to the smoke pipe. Owing to practical difficulties in keeping them in good condition superheaters decayed rapidly and were discarded for naval ships; to be revived in a few recent cases.

Some types of tubulous boilers, like the Belleville and Durr boilers, have superheaters, but these are practically steam driers only.

Another method of superheating in some tubulous boilers, although not primarily adopted for that purpose, is that of reducing the pressure in the steam pipe up to the engine much below that in the boiler. By this throttling or wire-drawing of the steam, no change is made in the *amount* of heat contained in a given quantity of boiler steam, which is allowed to flow into the steam pipe, but a different *distribution* is made which changes the nature of the steam. Take the case in which the boiler pressure of 315 pounds absolute is reduced to 265 pounds in the engine, and suppose that the boiler is supplying dry, saturated steam. The total heat of one pound of this steam at a pressure of 315 pounds is, from Table XV, 1210.6 units. But we know that to keep a pound of steam saturated and dry under a pressure of 265 pounds requires 1205.8 units. The difference, 4.8 units, has, therefore, gone to raise the temperature of the steam above that due to its pressure, or, to superheat it. If the boiler had not furnished dry but moist steam, the released heat would have first evaporated the moisture and made the steam dry, and then, if any heat remained, superheated it.

It will be noticed that in the case just mentioned the steam was allowed to expand freely in the steam pipe, where no work was done. There was, therefore, no liquefaction, due to conversion of heat into work, but there was a release of heat which became available for drying or superheating the steam.

A similar release of heat takes place, with similar results, when the final pressure in the H. P. or I. P. cylinder is greater, as it is generally, than the pressure in the corresponding receiver. The exhaust steam, therefore, expands freely in the receiver until its pressure is lowered to that of the receiver. As this exhaust steam is generally moist, the heat released is not more than sufficient to dry the steam, so that the superheating does not occur here, as a rule.

Compression or Cushioning.—If the exhaust port is closed before the piston arrives at the end of the stroke, the steam remaining in

cylinder is compressed into the clearance spaces, raising the pressure and consequently the temperature. The work done in compression is converted into heat, thus drying the steam, heating the cylinder walls and reducing to a slight extent the loss from cylinder condensation, which is made very appreciable when compression is carried up to the initial pressure. The great waste from back pressure is decreased from the earlier closing of the exhaust, and the cylinder not being open to the condenser, less time is allowed for re-evaporation. In addition to increasing the efficiency of the steam in this manner, cushioning has the mechanical advantages of checking the momentum of the moving parts gradually, and the direction of motion is reversed without the shock that would otherwise ensue.

Speed of Rotation.—High speeds have been found experimentally to give very decided gains compared with low speeds, the weight of steam used per hour per I. H. P. becoming smaller as the speed increases. On this account, together with the increased power obtained, the speed has steadily increased since the early days of the steam engine.

Compounding the Cylinders.—Of the different means mentioned for reducing cylinder condensation, the most important in general use is the multiple or stage expansion engine, whereby the exhaust steam from the high pressure or smallest of a series of cylinders is sent into a second or third and often a fourth cylinder before discharging into the condenser. As previously stated, the amount of liquefaction which takes place in a steam cylinder varies with the difference between the initial and final temperatures, so that as the pressure and number of expansions increase, the range of temperature also increases. In a single expansion engine this entire range is carried out in a single cylinder which is at one moment open to the boiler and the next in communication with the condenser for nearly the entire stroke. In the multiple expansion engine, the total expansion is divided into several stages and each stage carried out in a separate cylinder, thus reducing the range of pressure and temperature in each, and consequently the amount of cylinder condensation. For instance, steam of 165 pounds absolute and a temperature of 365° F. and

expanded down to 10 pounds with a temperature of 193.2° F., then exhausted into the condenser where the pressure is 3 pounds absolute and temperature 141° F., the range between the initial and final temperatures becomes 224° F. In a triple-expansion engine under the same initial and final pressures the high-pressure cylinder would work between 165 and about 55 pounds and the range of temperature would be 79° F. In the intermediate the range of pressure would be from 55 to about 18 pounds and a temperature range of 65° F., while with the low pressure the range would be from 18 pounds to 3 pounds pressure or temperature range of 80° F. Hence, under the latter condition, with the division in the expansion and range of temperatures among the cylinders, a much more economical engine is obtained.

Again, although the condensation takes place in each cylinder, it is not wholly a waste, as the water due to initial condensation and re-evaporation during the exhaust of one cylinder enters the succeeding cylinder as *steam* to do useful work on the piston of that cylinder. The only absolute waste occurring is that during the exhaust of the low-pressure cylinder, and this is greatly reduced owing to the much smaller range of temperature.

Efficiency of the steam is still further increased by stage expansion, owing to the reduction in clearance spaces. *Clearance volume* is the space enclosed between the piston, at the beginning of the stroke, and the face of the admission valve. This space must be filled with steam at each stroke and does not work upon the piston. This clearance volume is much smaller in the high-pressure cylinder of a stage expansion engine than it would be if the entire expansion had taken place in a single engine. Hence the loss of efficiency is much less.

On the other hand, the stage expansion engine possesses two losses in connection with the use of steam, which do not occur in the simple engine. One is the loss of pressure arising from friction in the passages between the cylinders, which may be reduced by properly proportioning these passages; the other is the loss from "drop" arising from the unresisted expansion of the exhaust steam from one cylinder into the receiver or space between that cylinder and the succeeding one. The steam of the receiver being of

a less pressure than the terminal pressure of the cylinder exhausting into it, expands in this space without the *performance of work* and with consequent loss. When this occurs the steam becomes to some extent superheated, and although this waste is not completely recovered, it is reduced by the advantage the succeeding cylinders obtain in expansion with superheated steam. Both of these steam losses are so much smaller than the gains mentioned that to secure the greatest advantage from high pressures, the use of multiple expansion becomes a necessity.

Since at the end of expansion the steam used occupies the whole volume of the low-pressure cylinder, if the loss from drop and friction between cylinders is disregarded the size of this cylinder for a given power should be the same as that of a simple engine developing the same power and working with the same pressure and ratio of expansion, so that the high-pressure cylinder does not add to the power; in fact, if cylinders were perfectly non-conducting, with no liquefaction, nor any losses between the cylinders, it would from a *thermodynamic* point of view be a matter of indifference whether expansion is carried out in one or three cylinders.

As in the case of the simple engine, a limit to the economical expansion of steam and useful steam pressures is reached in the compound or double, triple and quadruple expansion engines. Prof. Thurston gives the following total expansion ratio and pressure ranges for multiple-cylinder engines:

Steam expanded in	1	2	3	4 stages
Ratio of expansion...	2.5-3	6.25-9	16-27	40-81
Steam pressures, lbs.	25-30	60-100	120-300	350-800

In the foregoing the *efficiency of the steam* only has been considered in showing the advantages of the stage expansion engine, but, as will appear later, the *mechanical advantages*, due to a more equal distribution of stresses in the engine, especially for marine purposes, are equally valuable.

The economy in fuel of the triple over the double-expansion engine may be taken at from 20 to 30 per cent, the former using steam of from 150 to 160 pounds pressure and the latter steam of 90 pounds pressure. The coal consumption of well-designed mod-

ern triple-expansion engines varies from $1\frac{1}{4}$ to $1\frac{3}{4}$ pounds of good fuel per I. H. P. per hour.

The economy of an engine is also expressed in terms of the steam or feed water used by the engine per hour per I. H. P. The number of pounds of steam is obtained by measurement of the quantity of feed water pumped into the boiler per hour and evaporated, with the necessary allowance for leakage and withdrawal, or by measurement of the actual amount of condensed steam per hour taken from the condenser. With well-designed triple-expansion marine engines in good condition, in general, an economy of from 14 to 15 pounds of water per I. H. P. may be expected, while in the trials of the *Iona*, the machinery of which was especially designed with a view to an efficient and economical performance, an economy was obtained of 13.37 pounds of water per hour per I. H. P.

Indicated Horse Power.—The power developed by an engine, as determined by the use of the indicator, is called the indicated horse power.

Let P = mean effective pressure upon the piston in lbs per sq. in.

L = length of stroke of piston in *feet*.

A = area of piston in *inches*.

N = number of *revolutions* per minute.

Since the total load upon the piston = $P \times A$, and the distance moved through, equals the piston speed in feet per minute = $2 \times N \times L$ (since there are two strokes per revolution), the total work done per minute = $2 PLAN$., and as one horse power equals 33000 foot pounds per minute, we have I. H. P. = $\frac{2PLAN}{33000}$ (see Chapter X).

NOTE.—Quadruple expansion engines are not now installed in naval vessels; in ships steaming usually at designed speed, as in merchant ships, they are used. Naval ships cruise usually at reduced power, and under such conditions the loss in efficiency with quadruple expansion engines is too great to be tolerated.

CHAPTER III.

DESCRIPTION AND NOMENCLATURE OF NAVAL ENGINES. THROTTLE VALVES.

The steam after leaving the boiler through the main steam pipe passes through the engine-room stop valve, separator, and thence through a short section of steam piping to the H. P. valve chest. The admission of steam is controlled by means of a throttle valve bolted to the H. P. cylinder casing.

One of the requirements of a throttle valve is to provide for quick control in starting and stopping the engine, and necessitates some method of balancing the valve, especially if worked by hand gear.

On U. S. naval vessels two types of throttle valves are employed, the specifications requiring the valve to be balanced either by a piston or by being of the double poppet form. If a piston is used, it is required to work in a cylinder outside of the valve casing, steam being admitted to the back of the piston by means of a by-pass pipe and valve.

Double-Poppet Throttle Valve.—As fitted on the Illinois this valve is shown in Fig. 7, and consists of two disc valves, *A* and *B*, rigidly connected together, and secured to the valve stem by a collar and nut. Connection is made at the lower end of the valve stem to a system of levers, by means of which the valve can be opened and closed from the engine room platform. The two valves are slightly different in diameter, in this case the upper one *B* being the larger. The steam entering at *O* presses downward on *A* and upward on *B*, so that the valve is balanced, requires but little power to move it by hand, and can be opened and closed quickly.

The rectangular handhole *C* in the back of the valve chamber permits examination of both valves. The valve *B* may be removed when desired for repairs through this opening, while for the

removal of *A* the lower bonnet must first be removed. *P'-P* are openings for an outside by-pass pipe and valve connecting the two sides of the chamber. The opening *R* is fitted for a pipe connection to the receivers. Steam by this means can be admitted to the receivers by opening the by-pass *P* with the throttle closed in case of difficulty in starting.

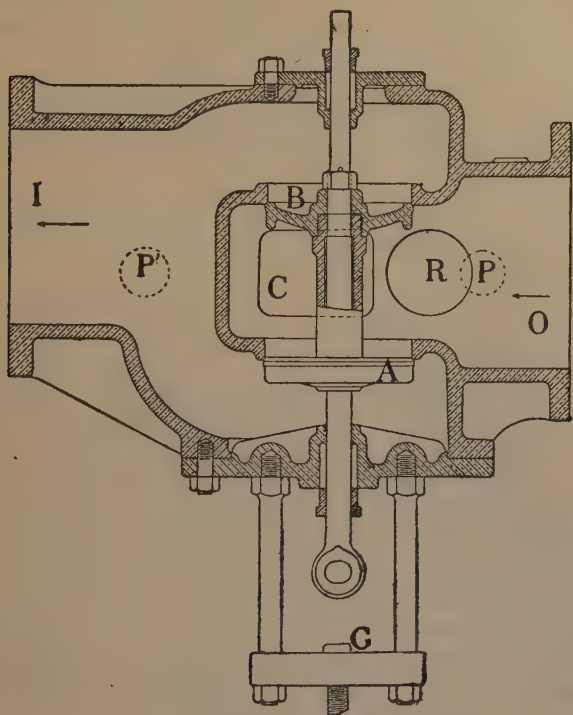


FIG. 7.

Piston Throttle Valve with Expansion Joint.—This valve, shown in Fig. 8, is fitted with a leaky balance piston *A*; the pressure at *B* is from the main steam line which has a stop valve next to the forward engine room bulkhead for closing off all steam to the main engine. Steam in the pipe *B* passes through the leaky piston *A* and through the by-pass pipe *C* to the lower part of the throttle or entrance to the H. P. valve chest, so that pressures in chambers *D* and *E* are always alike. When valve *F* begins to open the steam

pressure in *E* rises and this same pressure is communicated to the top chamber *D* through the by-pass pipe *C*. In order to prevent steam which leaks past piston *A* from entering the steam chest when

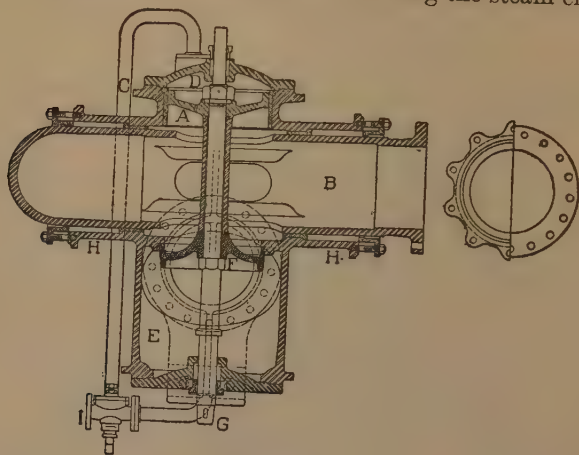


FIG. 8.

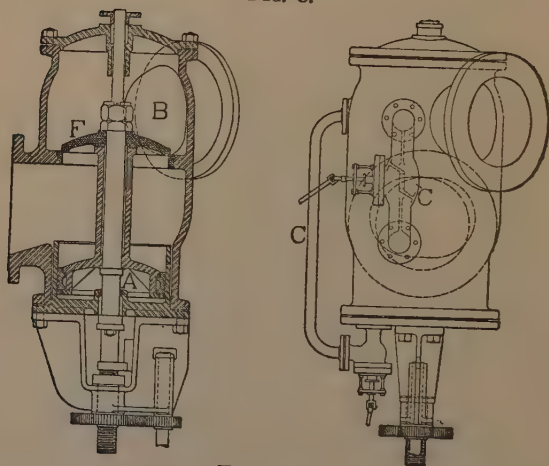


FIG. 8-A.

the engine is stopped and the stop valve not closed a 3-way cock *I* can be fitted in pipe *C* so as to direct steam from chamber *D* to the feed tank or hot well. In opening the throttle the 3-way cock can be so adjusted that pressure in the chamber *D* is just sufficient to

overcome the resistance to opening, and the operator will need to give only a very slight pull on the hand wheel, which is connected to *G*, to operate the valve.

Note the expansion joint with stuffing boxes *HH* on either side of the throttle valve. This allows for a slight movement of the main steam pipe as it expands and contracts, while permitting the throttle valve chamber proper to be firmly bolted to the steam chest.

Another form of throttle valve is shown in Fig. 8-A, having simply a balance piston, and by-passes *C* permitting steam entering at *B* to be admitted to the chamber *A*, thus allowing the pressure on the back of valve *F* to be overcome and the valve to open readily.

General Description of Propelling Engines.—In the following description of naval engines the type selected for illustration is that of the armored cruisers Tennessee and Washington, illustrated in Plates 1 and 2. The different parts are simply given by name and will be treated in detail in the succeeding chapters. The engines are of the vertical, inverted-cylinder, direct-acting, triple-expansion type, placed in water-tight compartments and separated by a middle line bulkhead. As the size of a single low-pressure cylinder for the required power would be inconveniently large, the power is divided between two low-pressure cylinders whose combined volume is equal to that of the single cylinder; and to assist in better balancing, and in the reduction of vibration, they are placed at the ends. The order of the cylinders, beginning forward, is forward L. P., H. P., I. P., and after L. P.; the forward L. P. and H. P. cranks are opposite, also the intermediate and after L. P. cranks, the second pair being at right angles with the first. The sequence of cranks then is H. P., I. P., forward L. P. and after L. P.

Referring to the vertical section through the center of the engine, steam on being admitted by the throttle valve enters the middle space of the *H. P. valve chest*, as the distributing valve within the *valve chest* in this case is an *inside valve* whose action will be explained in the succeeding chapter. This valve, termed the *H. P. valve*, is of the cylindrical or piston type, and is the type fitted to the H. P. and I. P. cylinders of engines using high-pressure steam. The *piston valve* is connected to the *valve stem* which

passes through a stuffing box at the lower end, and is connected to the valve gear and eccentrics, from which it derives an up-and-down motion, thus causing the valve to alternately open and close the *steam ports* giving communication through the passages 54 and 55 to the ends of the cylinder.

By thus admitting steam alternately to the upper and lower ends of the cylinder, the steam piston is driven up and down, this reciprocating rectilinear motion being converted into the circular motion of the crank shaft through the agency of the *connecting rod* and *crank*.

After the steam has utilized its energy in driving the piston, it is exhausted at the end of the stroke, the valve having moved so as to admit live steam at the opposite end, while the other end is exhausting through the same passages and ports by which it entered, but passing out by the opposite edges of the valve into the exhaust chamber, which in this case is at the end. This space is connected to the I. P. *receiver pipe*. By the term *receiver* is meant all the space between the piston at the end of its stroke and the valve of the succeeding cylinder.

The exhaust steam from the H. P. cylinder passes from the receiver pipe into the valve chest of the I. P. cylinder, where it acts as admission steam for that cylinder. Owing to its expansion in the H. P. cylinder and receiver, the pressure is less and volume greater than on admission to the H. P. valve chest.

From the increased size of the I. P. cylinder a larger valve is required, which in this case is obtained by two piston valves side by side, similar in their parts and acting in the same manner as a single large valve. The valve stems of both are connected to a yoke and driven by the same eccentric.

After the steam has performed work on the I. P. piston it is exhausted into the L. P. receiver pipes. These consist of two pipes leading from the exhaust space of the I. P. valve chest, one section forward, to the forward L. P. valve chest, and the other aft, to the after L. P. chest, where the steam is distributed to the cylinders in the same manner as before, by the aid of two distributing piston valves. From the L. P. cylinders the exhaust steam passes into the exhaust cavity in each valve chest and from there is carried through the *main exhaust pipe* to the *condenser*.

The reciprocating motion of the valve is given by an *eccentric* secured to the revolving crank shaft and transmitted by means of the valve gear, consisting in this engine of the *Stephenson Link Gear*. The *valve rod* is connected to a block sliding in a curved link whose extremities are connected to the ends of the eccentric rods of two eccentrics on the crank shaft. One eccentric is used for the ahead motion, while the other is fixed on the shaft in condition for motion astern. Only one eccentric at a time gives motion to the valve, the valve stem being brought in line with the *ahead*, or *backing* eccentric rod by sliding the link over the *link block* by means of the *reversing gear*. The link for each cylinder is connected up to a *reversing shaft* so that when the links are thrown over to the position desired they all move simultaneously.

The piston works steam tight within the cylinder and is secured to the upper end of the piston rod, and at the lower end of the piston rod connection is made with the *crosshead*, consisting of a rectangular block sliding upon the *crosshead guides or slides*. These guides are secured to the *engine frames*. Extending out longitudinally on either side of the body of the crosshead are two *crosshead pins* for the attachment of the *forked end* of the *connecting rod*, each branch being provided with a *bearing* and connections for one of the pins. The lower end of the connecting rod is connected to the crank pin by means of *brasses* and *cap*.

The cylinders are supported by the *engine frames* and consist in this engine of six front and back columns, each athwartship pair of columns being connected together by diagonal braces, and all tied together by fore-and-aft horizontal and diagonal tie rods as shown. The tops of the columns are flanged, faced and secured to the cylinder casing, and at the bottom to the *bed plate*. The bed plate consists of longitudinal and cross girders of cast steel, and is secured to the *engine keelsons* which are built up with the hull of the ship.

The *crank shaft* for the Tennessee and Washington is made in two sections, each section having two *cranks* and *coupling discs*, the sections are coupled to each other and secured by means of large coupling bolts and nuts.

The crank shaft is held in a number of *main bearings* fitted in

recesses in the cross girders of the bed plate. These bearings consist of two brasses, upper and lower, lined with white metal to reduce the friction and secured by the *main bearing bolts*. The *main journals* revolve within these brasses, a *journal* being the technical name given to that part of the revolving shaft inside the bearing. Two main bearings are usually fitted to each crank, placed as close to the crank arms as possible. Four of the six main bearings in Plates 1 and 2 are shown in section from which the details can be seen.

The crank is composed of two *crank arms* or *webs* and a *crank pin*, this latter forming a journal for the *crank-pin brasses* on the end of the connecting rod. The whole crank is in one piece with the shaft for naval engines, forged solid and afterwards machined to the shape and size. The space recessed in the engine frame under the crank is termed the *crank pit*.

A *reversing engine* is fulcrumed on the bed as shown in the elevation of the L. P. cylinder in Plates 1 and 2, and is used in moving the reversing shaft, thus throwing the links from one position to another much more quickly and conveniently than by hand power.

A *turning engine*, shown in the plate, is fitted to each main engine, so that the engines may be moved without steam on the main pistons. The engines must be moved in port to prevent sticking in any one position and for the work of repairs, adjustments, etc.

Recent Engine Design.—Plates 3 and 4 show the engines of the Oklahoma, one of the latest engined battleships. It is instructive to note the improvements in design over the engines of the Tennessee class, particularly the reduction of clearance volume effected by shortening and straightening the steam port passages.

CHAPTER IV.

THE SLIDE VALVE.

The distribution of steam in each cylinder of a marine engine for the process of admission, expansion, and final release either into the receiver spaces of succeeding cylinders, or to the exhaust passage leading to the condenser, is performed by means of the slide valve. The successful and economical results obtained from an engine depend to a great extent upon the design and adjustment of these valves, and to the skill and workmanship employed in their fitting.

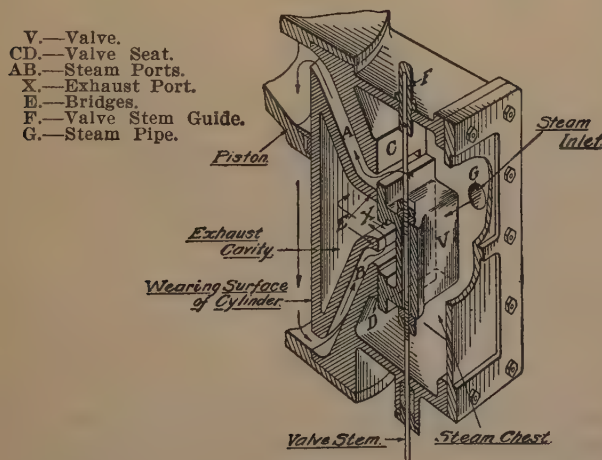


FIG. 9.

The action is the same whether a piston or flat slide is used, so that the latter in its simplest form, the single-ported flat slide valve, will be used to illustrate the movement of the valve in admission, cut-off, release and exhaust closure of the steam.

In Fig. 9, *V* is a single-ported slide valve contained within a steam chest filled with live steam. In the cylinder face or *valve seat CD* there are three rectangular openings called ports. *A* and *B* are the *steam ports*, one leading to each end of the cylinder; *X* is the *exhaust port* leading from the valve seat to the exhaust

cavity, as shown in the sketch, thence to the condenser, the receiver, or the atmosphere, depending upon the type of the engine. The face of the valve *V* is rectangular in shape, slides steam tight upon the valve seat and has a cavity on the under side which is always in communication with the exhaust opening, and alternately at certain portions of the stroke, open to one or the other of the steam ports, *A* and *B*, thus permitting steam to flow from either end into the exhaust passage.

In Fig. 10 the valve is shown in its *mid-position*, that is, in the middle of its travel, and it will be noticed that it not only covers the steam ports, but overlaps the edges to some extent at each

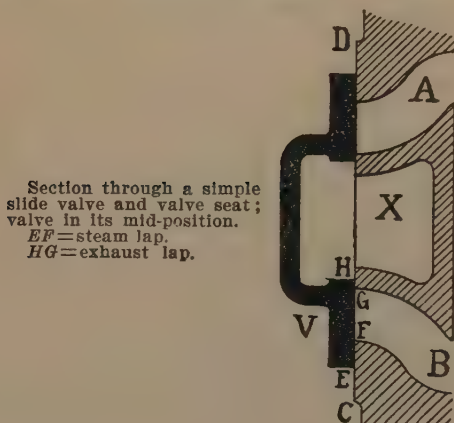


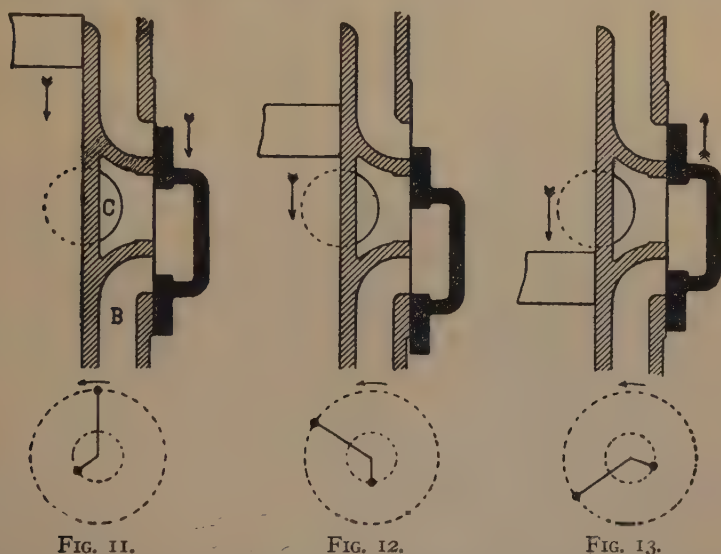
FIG. 10.

edge. The amount of this overlap on the steam side when the valve is in mid-position is known as the *steam lap*. The similar but shorter overlap on the exhaust edges is called *exhaust lap*. In this mid-position no steam can pass from the steam chest to either side of the piston, nor can any escape from the cylinder through either port into the exhaust.

The Movement of the Piston and Valve.—In analyzing the movement of the valve and distribution of steam during a complete revolution, assume the piston at the beginning of its stroke as in Fig. 11. For this position of the piston, the port should be uncovered by a determined amount. This amount of port opening

when the piston is at the commencement of the stroke is known as the *steam lead*, while the amount that the port on opposite end is open to exhaust, is known as the *exhaust lead*. The general *definition of lead*, then, is the amount by which the valve is open when the piston is at the beginning of its stroke.

Figs. 11 to 16 show a series of corresponding positions for the valve and piston during a single stroke of the engine or half revolution of the crank. In Fig. 11 live steam is entering the top end owing to the lead, and pushes the piston in the direction of the



arrow, the valve also having a movement in the same direction; at the same time steam on the opposite side of the piston is escaping through the port *B* and to the exhaust cavity *C*.

The piston and valve travel in this direction for a certain distance when the full port openings have been reached (Fig. 12), and the valve having reached the end of its travel in that direction, starts to return, gradually closing the port until, at the position shown in Fig. 13, the steam port at the upper end is closed, the admission of steam ceases, and this point is called the *point of cut-off*. From this point the piston is driven on by the expansive action of the steam contained behind the piston, while the lower

port still remains open to permit the escape of the exhaust steam, but is gradually closing.

The piston and valve continue to travel in opposite directions until they occupy the position shown in Fig. 14, the exhaust edge of the valve has covered the port at the bottom end and from this point on to the end of the stroke the steam remaining in the cylinder at the bottom end is compressed and its pressure will rise with the amount of compression. This is called the *point of exhaust*



FIG. 14.

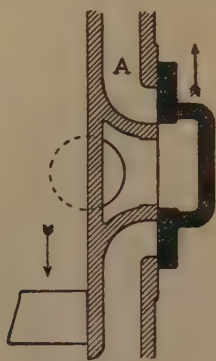


FIG. 15.

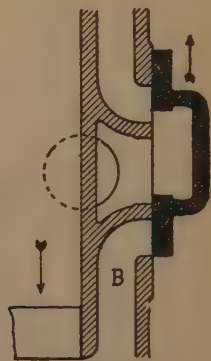


FIG. 16.

closure, or *point of compression*, and the operation, *compression* or *cushioning*. As will be shown later, compression acts as a cushion to absorb the momentum of the moving parts at the end of the stroke, bringing them to a gradual rest before the commencement of the next stroke and enabling them to pass over the crank center without shock; it also warms up the steam compressed.

The valve continues to move upward, when at a short distance further the upper end of the cylinder is opened to the exhaust (Fig. 15) by the inner edge of the valve, and the steam previously employed in pushing the piston ahead will now escape to the exhaust passage through the port A. This position is known as the *point of release*.

The valve still moving upward, the piston has nearly reached the end of its stroke, Fig. 16, the valve is just about to give admission to live steam at the lower end of the cylinder, through the port *B*, while the opening to the exhaust at the upper end is rapidly increasing. This position is called the *point of admission*, the valve gradually opening until when the piston has reached the bottom of its stroke the valve is open by an amount equal to the bottom lead. This completes the events for the down stroke. The return stroke is made with the same series for the piston going up, until the end of this latter stroke the valve and piston will occupy again the position shown in Fig. 11. During this process, two strokes of the engine have been made, and one complete revolution of the crank, and the operation will continue as long as steam is admitted to the engine.

Outside and Inside Valves.—In the preceding figures and descriptions of the movement of the slide valve, the steam enters by the outside edges and leaves by the inside edges, and is known as an *outside valve*. It is frequently convenient to reverse this condition, as, for instance, in one or more cylinders of a triple expansion engine, in which case steam is connected with the inside and the exhaust with the outside. Under these circumstances the valve is denoted as an *inside valve*, the steam lap being placed on the inside edges and the exhaust lap on the outside edges. It is clear from the action of the slide valve that while the outside valve moves with the piston at the beginning of the stroke and opposite to it during the latter part, the inside valve must act in the reverse manner, opposite to the piston at the beginning and in the same direction during the latter part of the stroke. In the triple expansion engine shown in Plates 1 and 2, the H. P. engine has an inside valve while the I. P. and L. P. have outside valves.

The Valve Motion and Eccentric.—The motion of the slide valve to accomplish the desired steam distribution is given by an *eccentric* and rod connected to the valve stem. The eccentric, Fig. 17, consists of a circular disc of metal *A*, called the *sheave*, keyed to the crank shaft, revolves with it, but whose center does not coincide with the center of the shaft. The center of the eccentric is at *P* while the center of the shaft is at *O*. Surrounding the eccentric

The weaker portion of the eccentric sheave is made of forged steel, as shown, to permit the sheave being smaller than if made of cast iron, while preserving the same eccentric radius.

See Plate 14 for details of eccentric sheave and strap.

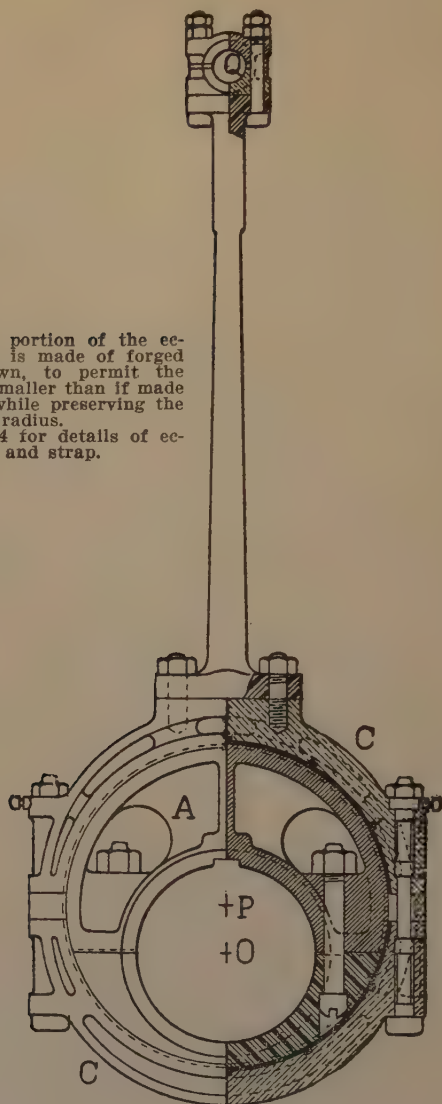


FIG. 17.

and moving on it are two semi-circular rings *C-C* forming the *eccentric strap*, bolted together and attached to the eccentric rod.

The other end of the rod is connected by a joint to the valve stem actuating the slide valve. In marine engines, as already explained,

this connection is made through a link. The strap has projecting edges working over the sheave so as not to slip off sideways.

As the crank shaft revolves the eccentric disc or sheave revolves with it and gives to the strap and eccentric rod a reciprocating motion, the same as though it were a crank OP attached to a connecting rod PQ , the point Q being constrained to move along a straight line OQ pointing to the shaft center. Practically, it is a crank of small throw but one with a crank pin enveloping the shaft, and of a diameter equal to the diameter of the sheave. The extent of the movement of Q , or the distance between its highest and lowest points, is evidently equal to twice the length OP . This distance OP is called the *eccentricity*, the *eccentric arm* or *eccentric radius*.

As shown in "Elements of Mechanism," the connecting rod introduces an inequality in the movement of the piston of an engine, so in the case of the eccentric, the point Q does not move with that regularity of change it would have, if the rod moved parallel with itself, instead of having an angular movement at Q as it moves up and down. The effect of this *angularity* of connecting rod is quite large in the case of the piston of an engine when the ratio of the lengths of crank and connecting rod is only from 4 to 6 to 1, but with an eccentric, the length of the rod is so great in proportion to the length of the eccentric arm that the deviation is so small that it is neglected, and the movement considered such as is made by a rod of infinite length. The point Q will move in a simple *harmonic motion* according to the following definition. If a point E' , Fig. 18, moves uniformly in a circle, the extremity e' of the perpendicular let fall from E' upon a fixed diameter has a simple harmonic motion.

Let $OE' = r$, $E'OB = \theta$ and v = linear velocity of E' ,

then $Oe' = r \cos. \theta$

Velocity of $e' = v \sin. \theta$

Acceleration of $e' = \frac{v^2}{r} \cos. \theta$

E' is termed the *generating point*

The angle θ is termed the *phase*

r is the *amplitude*.

The *period* is the time of one revolution of the point E' .

This movement of the point e' will measure the movement of the end of the eccentric rod and the valve to which it is attached. Under these conditions it will be seen that the valve is in its mid-position when the eccentric arm stands at right angles to the line of stroke, and it follows that the distance the valve has moved from its mid-position can always be obtained by multiplying the eccentric arm into the cosine of the angle made by the eccentric arm with the line of motion of the eccentric rod end.

Geometrical Representation of the Action of the Slide Valve.—

Assuming that the angularity of the eccentric rod has no appreciable effect upon the position of the valve, the following diagrams give the method of graphically representing the different crank positions for the various points in the distribution of steam to the engine. From these crank angles, and having given the length of

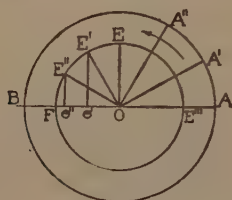


FIG. 18.

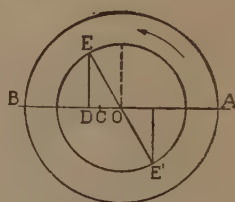


FIG. 19.

the connecting rod and crank, the corresponding piston position can be readily obtained.

Referring to Fig. 18, let AO represent the position of the crank on the dead center, and the valve to be without lap or lead. In this case the eccentric is set at right angles to the crank, or in the position OE . Assuming the eccentricity to be equal to OE , the travel of the valve will be $2 \times OE = E''F$. If the crank moves to the positions OA' , OA'' , etc., since the eccentric is secured to the shaft, it will also move through equal angular distances, and take the positions OE' , OE'' , etc. If from the points E' and E'' perpendiculars are let fall upon the diameter, the intercepts Oe' and Oe'' will represent the distance the valve has moved from its central position when the eccentric occupies the positions OE' and OE'' .

If the valve has lap and it is also desired that the steam port

may be open at the beginning of the stroke, i. e., the valve to have lead, the eccentric must be advanced on the shaft so that the angle between the crank and eccentric arm is greater than a right angle. In Fig. 19, lay off OC equal to the lap and CD equal to the lead. When the crank is on the center A the valve has been moved to the left a distance equal to OC plus CD . From D erect the perpendicular DE and join OE which gives the position of the eccentric at the commencement of the stroke. The angular distance through which the eccentric is moved in order to shift the valve from its mid-position to that which it should occupy at the beginning of the stroke is called the *angular advance* of the eccentric.

With outside valves the angular advance is laid off as shown, ahead of the crank, but with inside valves the lap plus lead is laid off towards the crank and the position of the eccentric arm OE' drawn as shown, behind the crank, directly opposite the position for an outside valve.

Instead of the valve having a direct connection to the eccentric rod, as assumed, if it is worked through a rocker arm or lever the location of the eccentric is the reverse of that given in the preceding. From these conditions and an inspection of the connections, the direction in which the engine will turn can be readily determined.

The Zeuner Valve Diagram.—Numerous forms of valve diagrams are employed to show at once the movements of the valve and the various events occurring in a stroke of the piston. The following method devised by Dr. Zeuner is probably the simplest and most comprehensive of those in use.

In Fig. 20, let P_3DE_3 be a circle whose diameter equals full travel of the valve in one direction, the crank is at the right hand center in the position OC , the angle of advance is DOP and the eccentric arm for this crank position is consequently at OP ; the valve having traveled from its central position the distance OA . If the crank move to positions C_1, C_2, C_3 , the eccentric moves through equal angles to positions P_1, P_2, P_3, P_3 being the position of the eccentric when the valve has reached its full travel to the left. It is evident that the crank at the same time has reached a position OC_3 , such that the angle $DOE_3 =$ angle of advance DOP .

a diameter, a circle be drawn, then a line drawn from the center O of the eccentric circle to any position of the crank, such as C_2 will intercept a distance OE_2 equal to the travel of the valve from its mid-position.

If a valve has neither lap nor lead and consequently the angle of advance zero, the diameter of the valve circle will be perpendicular to the line of dead centers of the crank. At the beginning of the stroke such a valve will be in mid-position, will complete its travel in one direction at half stroke and return to mid-position again at the end of the stroke.

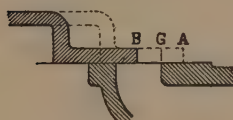
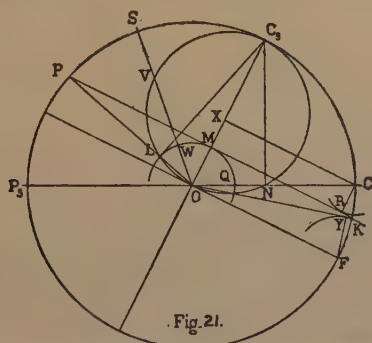


Fig. 22.

The Opening of the Steam Port.—Before the steam port can open, the valve must travel a distance equal to the lap. After it has opened, the amount of the opening will always be equal to the travel from mid-position minus the lap. In Fig. 22, let the dotted lines represent the valve in its middle position, the full lines its position at any given instant and AG the steam lap. Then AB is the distance the valve has traveled from its mid-position; hence,

Port opening $BG = AB - AG =$ distance of the valve from mid-position minus steam lap.

If, therefore, on the diagram, Fig. 21, a circle be drawn with radius $OM =$ steam lap, the port opening for any position of the

crank S will be the length of the intercept VW between the lap and valve circles. At crank position K this gives the port opening = zero, K is therefore the point where the valve begins to open, and gives the *point of admission*. At P the opening is again zero, the steam valve is again closed, and this crank position gives the *point of cut-off*.

Before applying the construction to a complete Zeuner diagram, certain geometrical properties used in the method will be explained.

1. A perpendicular let fall from the point on the eccentric circle corresponding to the angle of advance cuts the line of dead centers at a distance from the center = lap + lead.

This is evident, since the eccentric position must move a distance equal to the lap and lead in order that the valve may be correctly opened at the beginning of the stroke.

2. A line joining the points of admission and cut-off is tangent to the steam lap circle.

In Fig. 21, the triangles C_3OL and POM have the sides PO and MO equal, respectively, to C_3O and LO and the angle C_3OP is common. The triangles are therefore equal and the angle PMO = the angle C_3LO , which is a right angle, PM is therefore tangent to the steam lap circle.

3. If with center at the dead point C , and radius equal to the lead of the valve, a circle be drawn, it will be tangent to the line KP , joining the points of admission and cut-off.

Through C draw CX parallel to KP and therefore perpendicular to C_3O , the lines CO and C_3O are equal; angles OCX and OC_3N are equal; and the angle COC_3 is common to both triangles. Therefore the triangles COX and C_3ON are equal, $XO = NO$ and $XM = NQ$. But NQ = the lead. Therefore, $CR = XM$, the lead.

4. If OF be drawn perpendicular to OC_3 , a perpendicular from F on the admission line will be equal to the steam lap.

It will readily be seen that the triangles FOY and OKM are equal; therefore, $FY = OM$.

This property is made use of in constructing the exhaust part of the diagrams, whenever the lap circle is so small that it does not give a sharp intersection with the valve circle.

The foregoing refers to the steam side of the valve particularly.

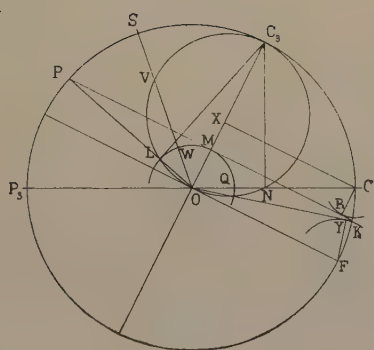


FIG. 21.

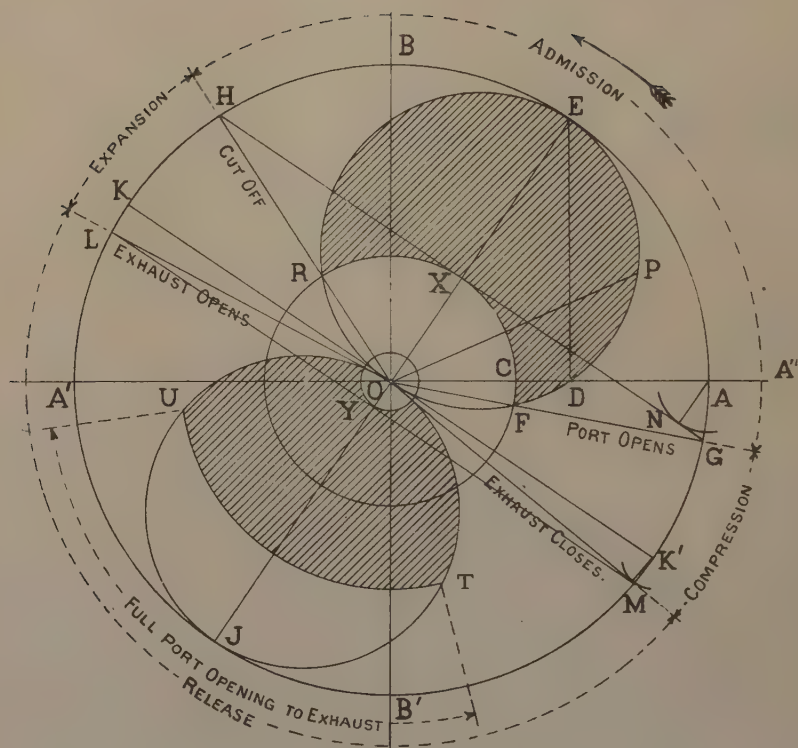


FIG. 23.

The exhaust is considered in the same way; but, with the piston making the return stroke from P_3 to C , Fig. 21, the diagram then comes in the lower half of the figure. The angle of advance and travel of the valve are the same, but the lap is the exhaust lap. The exhaust lead and the points of release and compression correspond respectively to the steam lead and the points of admission and cut-off, and are obtained in the same way.

The Complete Zeuner Diagram.—To apply the above principles to the construction of a complete diagram showing the functions of the valve for one side of the piston and for any position of the crank, we proceed as in the following:

With a radius OA equal to the eccentricity or length of eccentric arm, describe a circle, Fig. 23, and draw the horizontal center line $A-A'$ and vertical line $B-B'$. Let the crank occupy the position OA'' . From O lay off OC equal to the steam lap and CD equal to the steam lead and from D erect the perpendicular DE cutting the circumference at E . On OE as a diameter describe the steam valve circle OPE . With the center O and radius OC equal to the steam lap, describe the steam lap circle FCR . As before, chords from the point O to the circumference of the valve circle give the travel of valve from mid-position corresponding to crank positions given by these chords, and the valve opening is now shown graphically by the parts of these lines intercepted by the valve and lap circles.

Starting from the right of the figure the lap circle intersects the valve circle at F , the chord is common to both circles, the valve has moved to the left an amount equal to the lap and consequently the valve is just about to open. Hence, at crank position OG steam is about to be admitted before the commencement of the stroke. When the crank reaches the beginning of the stroke, the port is opened by an amount equal to $OD - OC = CD =$ steam lead. As the crank continues its revolution the valve gradually opens wider until at the crank position OE , when the valve has reached the limit of its travel to the left, and the port is wide open, or as wide as will be permitted by the throw of the eccentric. From this point on, the valve moves to the right, gradually closing the port, until at crank position OH , the circles intersect again and the

valve is closed and the point of cut-off reached, the piston now being driven by the expansive action of the steam contained in the cylinder. The steam has now been traced from admission to cut-off and expansion, and the next series of events is in relation to the exhaust.

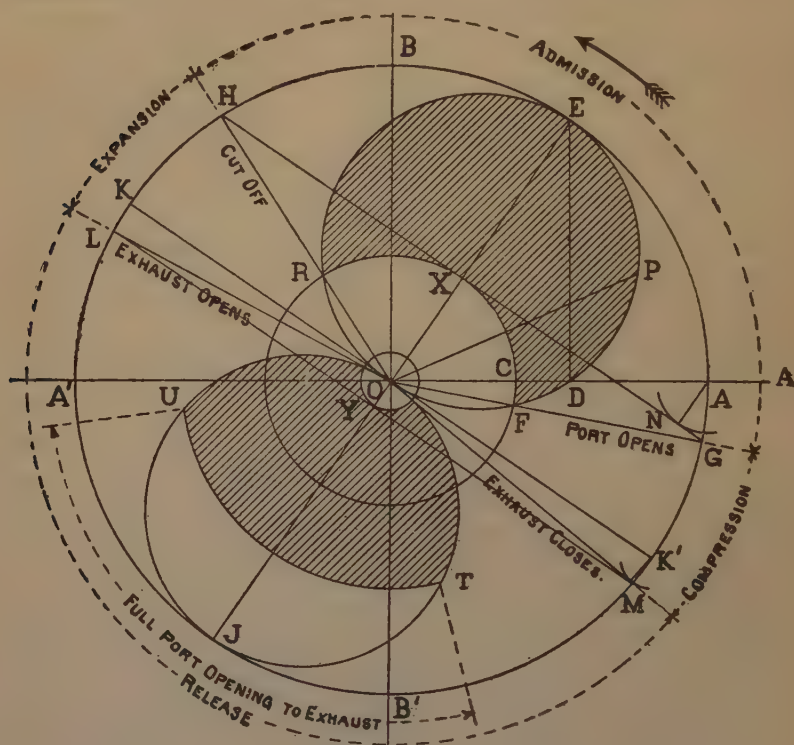


FIG. 23.

For the return (exhaust) stroke it is apparent that another valve circle may be drawn on OJ (a prolongation of OE) as a diameter and with the aid of which the exhaust movements may be traced as in the case of steam movements.

On OJ as a diameter draw the exhaust valve circle and from O as a center and radius equal to the exhaust lap describe the exhaust lap circle. At the crank position OK it does not intersect either valve circle and is at right angles to the line OE or OJ and the

valve is consequently in its middle position, the port still closed by the amount of the exhaust lap. The exhaust will not open until the valve travels from mid-position an amount equal to the exhaust lap; this occurs when the line of crank position OL cuts the intersection of the exhaust valve-circle and the exhaust lap-circle, and there the port is just about to open. The valve is still moving to the right, opening to exhaust wider and wider until at crank position OJ the furthest travel to the right has been reached. The valve is wide open and starts in the opposite direction towards closing. At crank position OM the exhaust lap and valve circles intersect and the valve is closed, and the remaining steam is compressed in the cylinder until the valve opens again to lead at OG .

The port is wide open to exhaust when the valve has traveled a distance from its mid-position equal to the width of the port plus exhaust lap, hence if an arc be drawn across the exhaust from the center O and radius equal to the width of port and exhaust lap, then for all crank positions between the intersections at U and T , the port will be wide open to exhaust.

Other useful geometrical facts besides those already mentioned may be observed from the diagram and will assist in the solution of problems connected with valve diagrams.

1. As the line joining cut-off and admission is tangent to the steam lap circle, so the line joining compression and release is tangent to the exhaust lap circle.

2. If AN is drawn perpendicular to GH , then $AN = CD =$ Lead.

3. The line KK' drawn perpendicular to EOJ will give the position of the crank when the valve is in mid-position.

4. A perpendicular from K' on the compression line OM will be equal to the exhaust lap. Use is made of this property for ascertaining more accurately the exhaust lap from the position of release, as it is sometimes difficult to determine the exact point of intersection of the compression line with the small exhaust circle.

In the diagram for the opposite side of the piston the *steam* circle for the top of the piston becomes the *exhaust* circle for the bottom, and vice versa. If the steam and exhaust lap are the same for the top and bottom the same circles would be continued

around the center to serve for both strokes. If they are different, as is usually the case in vertical engines, the correct radius must be used in each case.

Position of the Piston—Influence of Connecting Rod.—The influence of the length of the eccentric rod, as stated, can be neg-

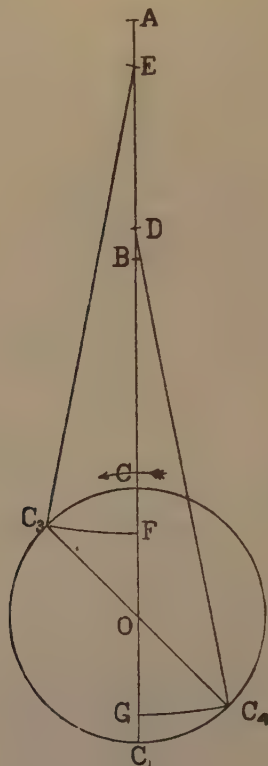


FIG. 24.

lected, owing to its great length compared with the eccentric arm. With the connecting rod and crank, however, the ratio is so much less that the distribution of steam is very unequal on the two strokes of a revolution.

To ascertain the variation, let C and C_1 , Fig. 24, be the dead centers of the crank OC turning in the direction shown and $C_3 C_4$

two positions of the crank making equal angles with the line of dead centers. The crosshead having the same motion as the piston, the position of the former will indicate that of the latter. From the crank position C_3 on the descending stroke the position of the crosshead is found by drawing from this point as a center and radius equal to the length of the connecting rod, an arc cutting the line of centers at E . Similarly, the crosshead position for the crank position C_4 on the up stroke is found at D , using C_4 as a center. From E and D as centers and radius equal to the length of connecting rod draw the arcs C_3F , and C_4G , then CF and C_1G are the distances the piston has moved from the beginning of each stroke, for $C_3E = FE = CA$, $\therefore CF = AE$.

From the figure it will be seen that on the down stroke steam is carried much further than on the up stroke and the power greatly increased. This is undesirable in vertical engines, as the weight of the reciprocating parts acts in the same direction assisting the steam in the down stroke and resisting in the up stroke. This inequality is adjusted by increasing the steam lap at the top end and decreasing it at the bottom, and by this means the effect of the descending weights, piston, rods, etc., is adjusted to suit the engine balance.

This equalization is also frequently accomplished by moving the valve a short distance along the rod towards the top end. The effect of this change is to make the point of cut-off later on the up stroke and earlier on the down stroke. The lead and exhaust lap at the bottom end are increased, thus giving a greater amount of cushioning, which counteracts to some extent the weight of the descending parts.

The crank pin circle can be drawn to any convenient scale from the center O and the valve diagram and crank diagram drawn together and the corresponding piston positions found as just described. By projecting these piston positions on a horizontal line of length equal to the diameter of the crank circle, and drawing the pressure and volume lines, the expected theoretical indicator diagram may be obtained, having given the initial steam pressure and vacuum. (See Plate 39.)

In Fig. 25, is given the valve diagram of the high pressure cylinder as designed for the engines of the Tennessee and Washington class of armored cruisers shown in Plates 1 and 2. The diagram is taken from a blue-print issued by the Bureau of Steam Engineering, and in the proposals for the construction of new vessels, the valve diagrams together with the data as designed (Table I), form one of a set of drawings to accompany the speci-

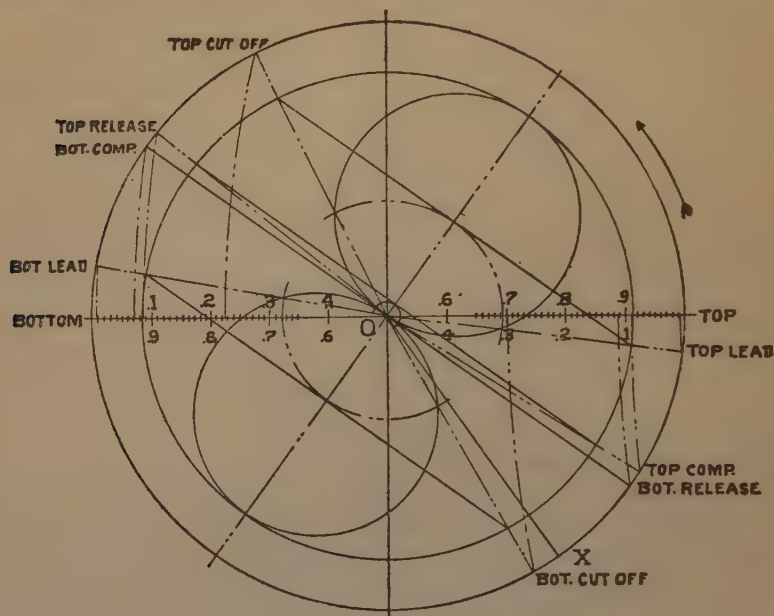


FIG. 25.

fications. As shown in the drawings of these engines the high pressure valve is an inside valve, the others, outside valves.

The eccentric is set at 180° or directly opposite the position for an outside valve under similar conditions, consequently it is behind the crank and at an angle from it equal to 90° minus angle of advance. OX is the position of the eccentric arm when the crank is on the top center. The center line for the valve circles is laid off as before, at an angle of 90° minus angle of advance, to the line of dead centers.

TABLE I.

NAME OF VESSEL.	ARMORED CRUISERS NOS. 10 AND 11.					
TOTAL I.H.P. AND DISPLACEMENT.	23000 I.H.P. 14500 TONS DISPLACEMENT.					
TYPE AND NO. OF ENGINES.	4 CYL. VERTICAL INVERTED TRIPLE ENG. TWIN SCREW					
DIAMS. OF CYLS. AND STROKE	38½", 63½", 74", 74" 48" STROKE.					
VALVE GEAR.	STEPHENSON LINK.					
CONNECTING ROD.	96" BETWEEN CENTERS.					
SCALE OF CRANK CIRCLE 3"=1 FT.		SCALE OF VALVE CIRCLE, FULL SIZE.				
	H.P.		I.P.		L.P.	
ECCENTRICITY.	5"		5"		5"	
TRAVEL OF VALVE.	10"		10"		10"	
KIND & NO. OF VALVES. DIA. IF PISTON.	ONE PIST. 24½"		TWO PIST. 27"		TWO PIST. 27"	
SIZE OF VALVE ON WHICH STEAM TAKEN	INSIDE.		OUTSIDE.		OUTSIDE.	
	TOP.	BOT.	TOP.	BOT.	TOP.	BOT.
WIDTH OF PORT & LENGTH IF SLIDE.	4½"	4½"	4½"	4½"	4½"	4½"
STEAM LAP.	2⅜	2⅛	2⅜	2	2⅜	2
EXHAUST LAP.	-⅝	+0	-⅝	+0	+⅙	+⅝
ANGULAR ADVANCE.	35°		37°		37°	
STEAM LEAD	ANGULAR.		7°	10°	10.75°	13.5°
	LINEAR.		½	¾	⅙	1
CUT OFF IN INCHES.	37⅜	33⅜	37⅜	33⅜	37⅜	33⅜
CUT OFF IN DECIMAL OF STROKE.	.775	.700	.780	.699	.780	.699
EXHAUST RELEASE IN INCHES.	4⅙	5⅙	4⅙	6	3⅙	3⅙
EXHAUST RELEASE IN DECIMALS.	.087	.111	.097	.125	.08	.083
COMPRESSION IN INCHES.	4⅙	3⅙	5⅙	3⅙	6⅙	5⅙
COMPRESSION IN DECIM. OF STROKE	.09	.07	.105	.08	.129	.113
STEAM OPENING.	2⅝	2⅝	2⅙	3	2⅙	3
EXHAUST OPENING.	FULL PORT.		FULL PORT.		FULL PORT.	
VELOCITY OF STEAM	960' PER MIN.		960' PER MIN.		960' PER MIN.	
IN FT. PER SEC. @.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.
VELOCITY OF EXHAUST	960' PER MIN.		960' PER MIN.		960' PER MIN.	
IN FT. PER SEC. @.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.	1120 R.P.M.
VELOCITY OF EXHAUST TO CONDENSER.	120 FT. PER SEC.		120 FT. PER SEC.		120 FT. PER SEC.	

It will be noticed in the data and results, Table I, that the exhaust lap is negative in the H. P. and I. P. valves, that is, when the valve is in mid-position, instead of the exhaust edges just covering the port or overlapping it, the edges are cut away slightly, leaving the port open to exhaust in this position. The amount by which the port is uncovered when the valve is in mid-position is called *negative lap*. Under the circumstances of negative lap, the intersection of the exhaust lap circle with the steam circle must be taken to obtain the positions at release and compression. The complete diagram, then, gives the following information.

1. Admission
2. Cut-off.
3. Steam lap
4. Steam lead
5. Maximum opening of steam port
6. Angle of advance
7. Travel of valve
8. Release of steam
9. Compression
10. Exhaust lap
11. Exhaust lead
12. Maximum opening of exhaust port.

Points 1, 2, 8 and 9 are given and are measured on valve diagram by the *angular* distance of the crank from beginning or end of the stroke. Frequently, these points are given as taking place at a certain part of the stroke. To find the angular distance corresponding to such data, the eccentric circle may be used on a proper scale to represent the crank circle, but the beginner should use a larger circle for this purpose, as in Fig. 25, to avoid confusion.

Points 1 to 7 belong to the steam side of the diagram and points 6 to 12 to the exhaust side, points 6 and 7 being common to both.

To construct a complete diagram it is necessary to have four of these points. Three are sufficient for the steam or exhaust side alone, but one or two of the points must belong to a different side from the others, to construct a complete diagram. One of the

points must be one of the linear measurements on the diagram, i. e., points 3, 4, 5, 7, 10, 11 or 12. This is seen from the fact that if only angles are given the linear dimensions of the valve could be made of various dimensions so long as they bear a certain ratio to each other.

The position of the piston for any crank position can also be represented graphically by the following method.

Referring to Fig. 26, draw AB and CD at right angles to each other. From the intersection O and with a radius OE equal to the radius of the crank to any convenient scale describe the crank-pin circle. With E as a center and radius ER_1 , equal to the length of the connecting rod, describe the circle R_1, R_2, R_3, R_4 .

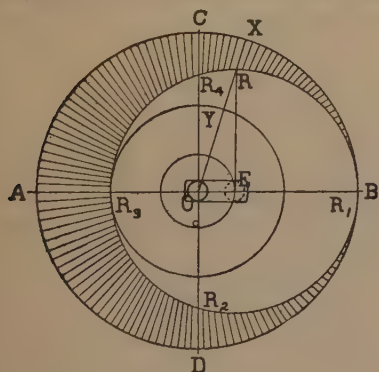


FIG. 26.

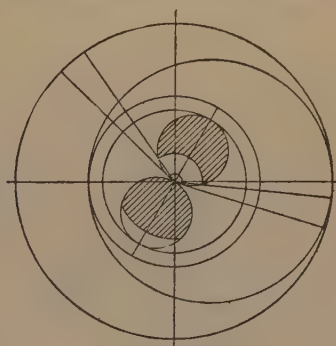


FIG. 27.

From the center of the shaft O and a radius OR_3 describe a circle and again with O as a center and radius OR_1 , describe the outer circle. For any crank position such as given by the direction OE , the piston has moved from its extreme position a distance equal to RX and is distant from the opposite end of the stroke by the distance RY , for supposing the piston connected directly to the crank pin as in a trunk engine and the crank fixed in the position OE instead of revolving, while the cylinder revolves around the shaft, then the path of the piston during the revolution will lie on the circumference R_1, R_2, R_3, R_4 , since it is always at the same distance ER from the crank pin. The top or outward end of the cylinder moves on the circumference $XBDA$; while the circle

whose radius is OK_3 represents the path of the bottom end of the cylinder. The inequality of the piston's motion during the up and down strokes owing to the influence of the connecting rod can be readily seen from the diagram.

The same reasoning is true if the cylinder is fixed at R_1 and the shaft made to revolve.

This diagram can be used with a Zeuner diagram by drawing it to a larger scale than the valve diagram so as to clear the latter, Fig. 27, and by this means the exact position of the piston for any position of the crank can be readily obtained.

PROBLEMS.

1. Given travel of valve and points of admission, cut-off and release.

Referring to Fig. 23; let $AA' =$ the given travel. On AA' as a diameter, construct the circle $ABA'B'$. G , H and L are the given points, found with the data given. Through G and H draw GH , through O draw EJ perpendicular to GH , and through L draw LM parallel to GH . X and Y are thus given and, by drawing the valve circles on OE and OJ , and the lap circles through X and Y , the leads are given.

2. Given travel of valve, steam lap, steam lead, and any point in the exhaust diagram.

Construct the circle $ABA'B'$ as before. Lay off $OD =$ lap + lead, and erect the perpendicular DE . EOB is the angle of advance. The remainder of the construction will be plain.

3. Given travel of valve, cut-off, steam lead, and exhaust lap.

Construct the circle $ABA'B'$ as before and mark the point H with the data given. With center A and radius $=$ the lead, draw a circle. Through H draw a line passing tangent to this circle below A . This cuts the large circle in G , the point of admission. Draw a diameter to the large circle perpendicular to HG . This gives the angle of advance and the remainder of the construction will be plain.

4. Given steam lap, steam lead, and the point of cut-off.

In Fig. 23, let AOH equal the angle corresponding to the point of cut-off. Draw the lap circle CR , and lay off $CD =$ the given

lead. Construct a circle passing through the three points D , O , and R . This will be the valve circle and its diameter = the half travel of the valve.

5. Given cut-off, release, compression, and width of ports.

The width of port is taken as equivalent to the maximum opening of port to exhaust (assumed in such problems, not always so).

Draw a circle of indefinite radius (Fig. 28), and mark on it the points of cut-off, release, and compression. Join the points of release and compression by the line AB and through O draw a diameter perpendicular to AB . Then:

$$\frac{\text{Travel of valve}}{\text{Given width of port}} = \frac{WY}{XY}.$$

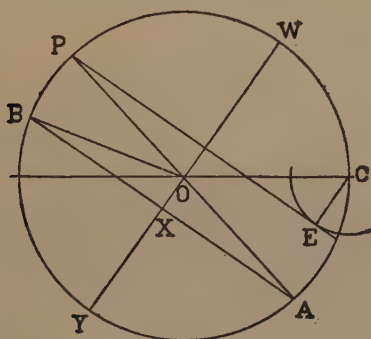


FIG. 28.

From this the travel of the valve is obtained and the diagram may be constructed. If the lead is given instead of the width of port, through P draw a line parallel to AB and from C let fall a perpendicular CE upon it. Then

$$\frac{\text{Travel of valve}}{\text{given lead}} = \frac{WY}{CE}.$$

6. The travel of a valve is 6". The cut-off occurs when the crank has completed 105° of its path. Admission of steam begins when the crank is within 7.5° of the beginning of its stroke. Exhaust closes when crank is 60° from the end of its stroke.

Construct the Zuener valve diagram and find:

Steam lap; steam lead; exhaust lap; exhaust lead; angle of advance.

7. Travel of the valve = 5". Steam lap = 1". Exhaust lap = $\frac{1}{2}$ ". Steam lead = $\frac{1}{2}$ ".

Find and measure the angle of advance.

If the stroke of the engine = 4' and the connecting rod be considered as infinite, how far is the piston from the end of its stroke when cut-off occurs?

8. What should be the steam lap of the valve in the preceding question so that the steam would be cut off at half stroke?

9. What angle of advance should be given the valve of the engine of question 7, so that it would cut off at half stroke?

How would this change affect the other functions of the valve?

10. Stroke of engine = 3'. Length of the connecting rod = 5'. The valve has an exhaust lap of $\frac{1}{4}$ ", and a steam lap of 1". The maximum port opening for exhaust is 2.75". Compression begins when the piston is 12" from the end of its stroke.

Find the travel of the valve and the angle of advance.

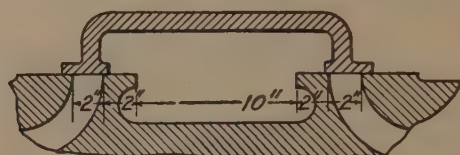


FIG. 29.

11. The valve, a sketch of which is shown, is to have a travel just sufficient to open the port wide for exhaust when the exhaust is a maximum.

With an infinite connecting rod, it is to cut off at .75 of the stroke of the engine.

The exhaust is to begin to open when the piston is at $\frac{1}{8}$ of its stroke from the end, and the admission to occur $\frac{1}{16}$ of the stroke from the end.

Find the steam and exhaust lap, the travel of the valve, the angle of advance of the eccentric, and the position of the piston at exhaust closure.

12. In a given engine the cut-off is to occur when the crank is within 45° of the end of its stroke, the release within 15° of the

end of the stroke, and the admission to begin 7.5° from the beginning of the stroke.

Maximum exhaust port opening $2.5''$.

Find the travel of the valve, steam and exhaust lap, crank angle when compression begins and the steam and exhaust leads.

13. Stroke of an engine $= 3'$. Length of its connecting rod $= 5'$.

When cut-off occurs, the piston is $2'$ from the beginning of the stroke. When release occurs, the piston is $5''$ from the end of the stroke, and when compression begins, at $10''$ from the end of stroke. The maximum opening of the port for steam is $1''$, and for exhaust $2''$.

Construct the diagram and measure the angular advance.

Make a section of the valve and its seat to a scale of $3'' = 1'$, and show the dimensions of all the parts.

14. Steam lap $= 2''$; exhaust lap $= 1''$; angle of advance $= 30^\circ$; cut-off at .75 of the stroke. Having an infinite connecting rod, construct the diagram.

This diagram being that of an engine with link in full gear, show how it would be affected by open rods; by crossed rods; and by raising the link in each case one-half.

If, instead of a link motion, a radial gear be used, and the cut-off be shortened by raising the reversing lever, show how the diagram would be affected.

15. Travel of the valve $= 6''$; cut-off at half stroke; angle of advance of the eccentric $= 30^\circ$; exhaust lap $= 1\frac{1}{2}''$. Construct the diagram.

(a) Construct the theoretical indicator card with an initial pressure of 100 lbs. to the square inch; the scale of the indicator being 25 lbs. $= 1''$, and the curve of expansion having the form $pv = \text{constant}$.

(b) If the angular advance be diminished, show how the diagram and the indicator card will be affected.

(c) What alteration in the data might be made so that an earlier exhaust closure would follow?

16. Steam lead $= \frac{1}{2}''$; steam lap $= 1''$; exhaust lead $= \frac{3}{4}''$; travel of valve $= 3''$.

takes the position OP' , making an angle θ with the line of dead centers equal to the angle of advance, or for a crank position OA , the eccentric position is OP' . From P' divide the eccentric circle into the same number of equal parts as $1', 2', 3'$, etc., as in the case of the crank circle. Drawing horizontal lines from these eccentric positions and vertical from the crank pin positions, the intersections of the corresponding lines when joined by a fair curve form a closed figure called the *valve ellipse*.

Influence of Oblliquity of Connecting and Eccentric Rods.—Fig. 31, shows the valve ellipse for the H. P. valve of the U. S. Armored Cruisers Washington and Tennessee, the data for which are given on page 69.

The larger circle is the crank pin circle, and the smaller the eccentric circle, divided exactly as in the last figure. To find the position of the piston corresponding to any crank-pin position a , take the length of the connecting rod as a radius and with a center on BA extended, strike the arc ab , b is the piston position required. Similarly, to find the true position of the valve, take the length of the eccentric rod (from center of eccentric to center of valve rod pin), as a radius, and with a center on the vertical center line extended, strike the arc cd , d is the position of the valve required. Projecting b vertically and d horizontally, e , the point of intersection, will be a point on a curve which gives the true relative motion of the piston and valve. This same operation must be followed to obtain the remaining points of the ellipse.

This diagram by means of a single curve shows the relative movement of both the valve and the piston, and is frequently used to show the effects due to varying the steam and exhaust lap and the angular advance.

In Fig. 31 let AB represent to scale the stroke of the piston, and ordinates on this line the displacement of the valve from mid-position; movements upward from mid-position being shown above AB , and all movements downward from mid-position, below AB . For an outside valve the different phases on the *up* stroke will be shown above the base line, and below that line for events on the *down* stroke. With an inside valve like the one under consideration this relation is reversed.

Let AF be set off from AB equal to the top steam lap and EF drawn parallel to the base line; since the exhaust lap is negative draw CD parallel to and a distance above the base line equal to the exhaust lap on the top end of the valve, as the action of this edge in opening and closing the exhaust is due to motion above the mid-position. Similarly, lay off parallel to and a distance below

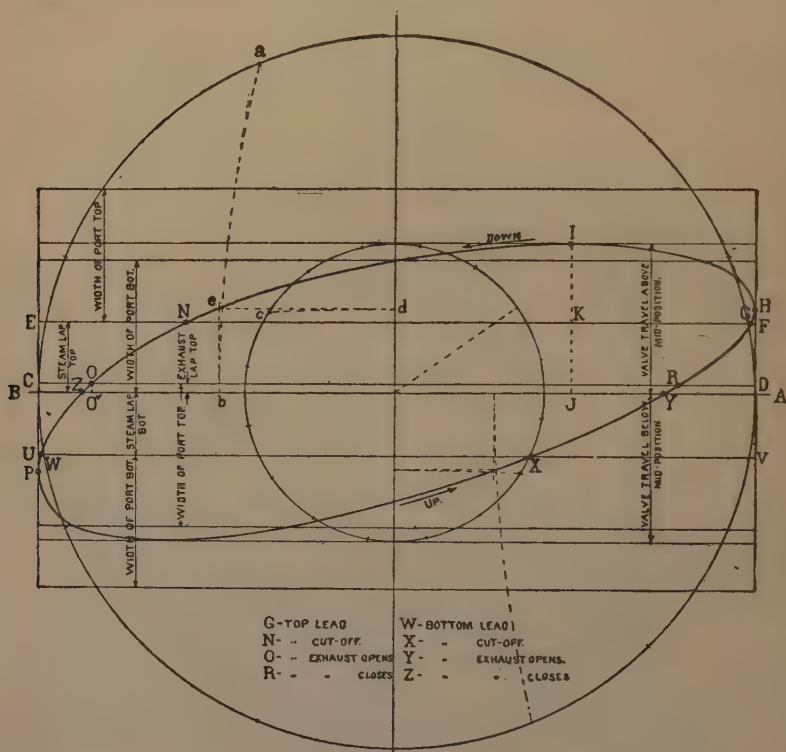


FIG. 31.

the base line, UV equal to the bottom steam lap. As the exhaust lap is zero for the bottom, the exhaust lap line will coincide with the center line AB . Then by drawing the valve ellipse as explained, the points of admission, cut-off, release and compression are determined by the intersections of the valve ellipse with the respective edges of the lap lines.

Commencing at the top and the piston moving down; since the valve starting from the mid-position must move a distance equal to the outside lap before the port commences to open and the curve represents the travel of the valve from mid-position, the distance between the lap line EF and the curve at any given point must represent the distance the steam edge of the valve has moved from the edge of the port. The curve cuts the lap line at G , a short distance from the end of the stroke, and the valve is consequently just opening to *lead*, the piston at the distance GF from the end. At the end or commencement of stroke, the valve has traveled a distance AH from the middle of its stroke and FH must be the lead. From this point the valve moves upward from mid-position until at I the highest point of the curve is reached with a piston position J , and the maximum port opening is reached, and IK represents the maximum port opening. At N the curve again cuts the lap line, showing the valve covering the port, and the steam is *cut off*. The port remains closed until the curve cuts the line of top exhaust lap at O , showing the inner edge of the valve is now opening and exhaust lead begins at piston distance of BO' from the end. The end of the stroke is reached at P , the valve having an exhaust lead equal to CP , and remains open during the return stroke when the curve again cuts the exhaust lap line at R the valve is closed and compression continues until the valve again opens to lead at F . Similarly steam can be taken from the opposite end and followed around in the same manner.

This diagram is not suitable when designing new valves, but for following out the action of gear already designed or in action it is very convenient.

With the aid of this diagram the effects of a change in the steam lap can be readily seen as it is obtained by merely raising or lowering the steam lap lines EF and UV . Raising the lap line or increasing the lap produces an earlier cut-off, reduces the lead and decreases the port opening throughout the travel. Reducing the lap produces the opposite result.

Similarly, by changing the position of the exhaust lap line for increased exhaust lap it is seen that the exhaust opens later, closes earlier and with decreased exhaust lead.

By increasing the angle of advance and laying off a new set of diagrams for comparison with the former one, the steam and exhaust leads will be found to be increased, and all functions of the valve *earlier* for an increased angle of advance. For a reduction of this angle the contrary will be clearly seen.

The valve ellipse may be drawn by the engine itself, by reducing the motion of the piston by a pantograph and transmitting it to a sheet of paper, against which is pressed a pencil whose motion

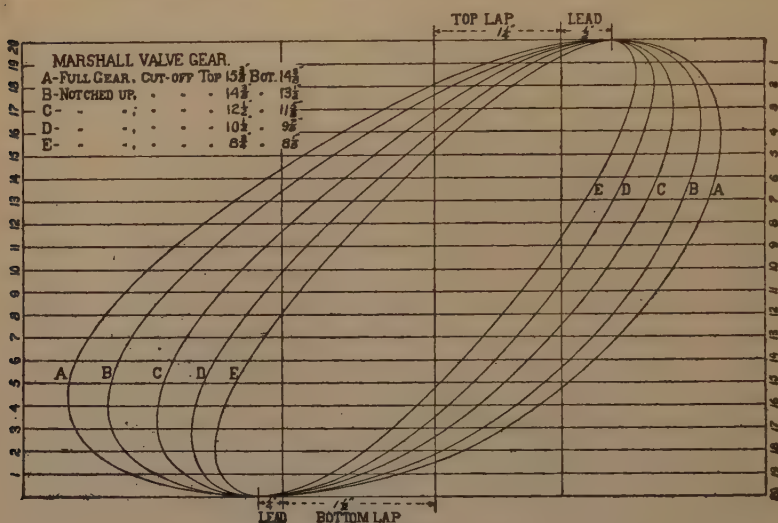


FIG. 32.

is taken from the valve gear. This diagram would be the same as that laid off to the same scale by the method already given.

In Fig. 32 is an elliptical diagram taken from the engines of a U. S. naval vessel fitted with a Marshall Valve Gear. From this diagram can be seen at a glance the relative movement of the valve and piston, the effect of "linking-in" on the events of the valve and the peculiarities of the Marshall Valve Gear, viz., constant lead for all points of cut-off, inequality of the port opening for the top and bottom stroke and the rapid opening and closing of the ports.

The disadvantage of this system of studying the motion of a valve is the necessity of constructing a curve.

Double-Ported Flat Slide Valves.—In the single-ported valve there is but one port for the steam to enter and leave at each end of the cylinder. For large cylinders, such as the L. P. cylinder of triple expansion engines, the ports become very large and the travel of the valve to open the ports sufficiently becomes inconveniently great. This involves the use of a large eccentric and a

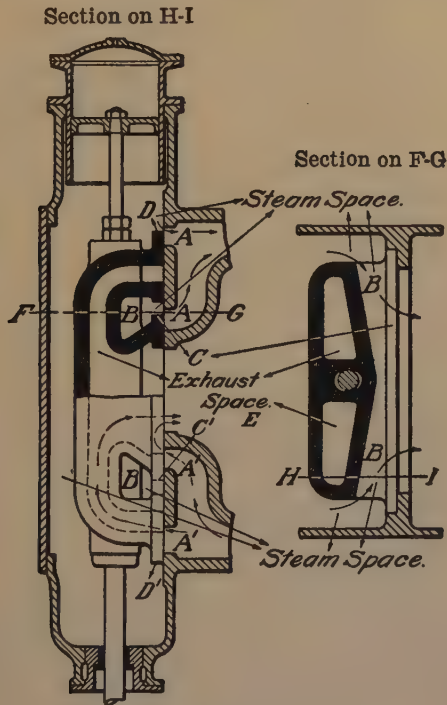


FIG. 33.

considerable expenditure of power to work it. To reduce the travel and thereby decrease the work done by the eccentric, a *double or triple-ported valve* is employed, the former being in more general use.

Referring to Fig. 33, which represents two sectional views of a double-ported slide valve, the steam passages AA'-A'A', leading to each end of the cylinder, instead of terminating at a single port

in the valve seat, are divided into two ports, each one-half the width necessary for the single-ported valve, so that for the required quantity of steam the valve need only move with one-half the travel required for the ordinary valve, for the same area of port opening.

Steam surrounds the valve and is admitted at the extreme end ports in the same manner as the single-ported valve; but in addition steam passes through openings in the side to the chambers *BB* through the body of the valve. The admission and egress of steam for the inner steam ports is controlled by the faces *CC'* which are fitted with the required lap exactly as are the outside valve faces *DD'*.

Valve is shown admitting steam at top. Connecting it up with the eccentric, the same succession of operations will occur as with the single-ported valve, both ports *A* and *A'* being opened and closed simultaneously. The valve being moved down, steam is admitted over the edges *D* and *C* while steam is exhausted from the opposite end of the cylinder through the ports *A'A'* into the cavity of the valve and out to the condenser through *E*. If three ports and three valve faces are used at each end, the valve is triple-ported and the travel is reduced two-thirds.

Balance or Equilibrium Ring.—In large engines fitted with flat slide valves the total pressure on the back of the valve presses it hard against the valve seat, involving a considerable expenditure of power, excessive wear on valve face and seat, together with a heavy load on the eccentric. To reduce this effect a balance ring is fitted at the back of the valve which largely reduces the area exposed to steam pressure and so reduces the load.

The ring is fitted steam tight between the back of the valve and the valve chest cover or bonnet, and is either carried on the back of the valve, pressed by springs against the inside of the cover, or is carried on the inside of the cover and pressed out against the back of the valve. In the latter case the ring is stationary, while in the former the ring moves with the valve. When the balance rings are fitted in the valve chest bonnet an advantage is gained by enabling the packing rings to be adjusted from time to time from the outside, whether the engines are working or idle. The space

within the ring is connected by piping with the exhaust. The area of the balancing space must be so arranged as to leave sufficient area exposed to steam to keep the valve on its seat.

The two methods of fitting equilibrium rings are shown in Figs. 34 and 35. The former is the method adopted for the flat slide valve on the L. P. cylinder of the Ohio. *W* is the space within the balancing ring. A grooved cast iron ring *R* is bolted to the back of the valve and carries the balancing ring *T*, the upper sur-

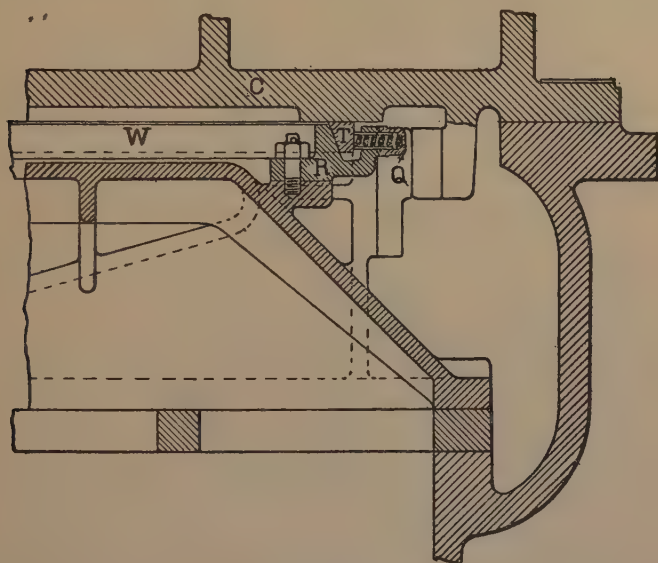


FIG. 34.

face of *T* making a steam-tight joint with the scraped surface of the under side of the valve chest bonnet *C*, by the steam pressure on the sides of *T*, and further by means of a spring set around the circumference of *T* and acting on the wedge-shaped surface between *T* and *R*. Composition caps *Q* resist the tension of the springs. Sliding bushings closed at one end guide the springs and prevent the direct contact of steam. The ring is supposed to be self-adjusting, the angle between *R* and *T* being made with this point in view.

Fig. 35 is one of the designs where the balancing ring is stationary, and similar to that used on the New Orleans. *A* is the equilibrium ring; a flexible copper ring of the section shown at *B* is connected to the equilibrium ring and valve chest cover *C*. Adjustments can be made from the outside by means of set screws *D*, passing through the cover and acting against a spring. A cock should be fitted to the pipe leading from the back of the valve and a cock to open the balancing space to the atmosphere. By this means any leakage past the ring may be discovered; a pressure gage is also fitted for the same purpose.

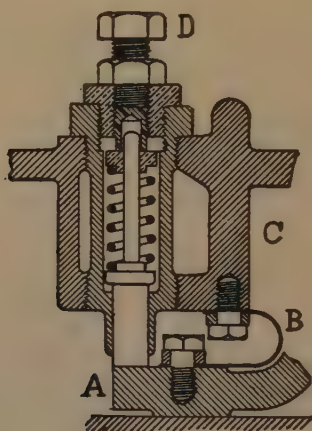


FIG. 35.

Balancing rings for the back of flat slide valves, however, do not work satisfactorily, they are hard to adjust and difficult to keep in adjustment, and for high pressures the flat faced valve is replaced by the piston valve.

Piston Valves.—To avoid the difficulty experienced with flat slide valves in the heavy frictional load to be overcome, the H. P. and I. P. and frequently the L. P. cylinders are fitted with cylindrical or piston valves. The piston valve is practically a flat slide valve wrapped into the form of a cylinder with the valve seat surrounding it. The steam does not cause any pressure between the rubbing surfaces; the only resistance to be overcome is the

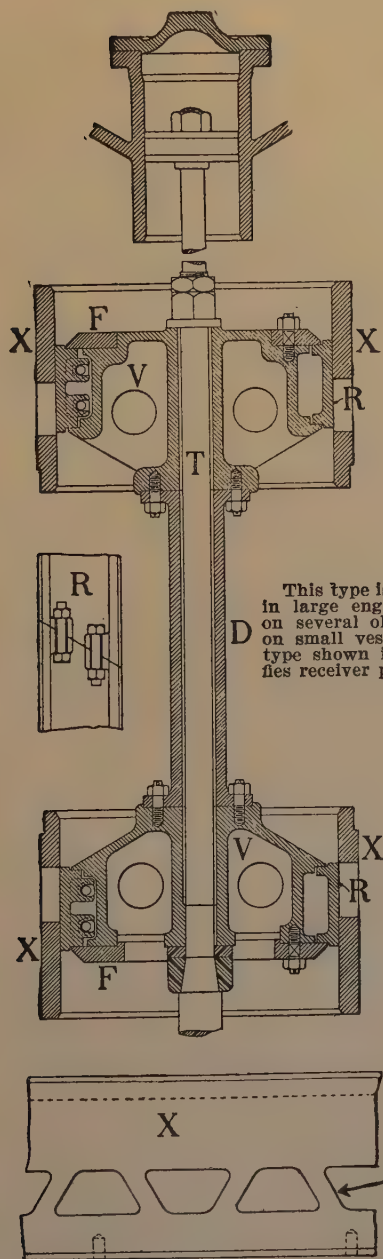


FIG. 36.

weight of the valve and such frictional resistance as is brought on by the spring packing ring fitted to insure tightness.

One type of the piston valve is represented in Fig. 36 and consists of two discs or pistons connected to an extension of the valve stem *T*. The pistons are held together rigidly and the correct distance apart by means of the distance piece *D*, and secured by a collar and nut on the outer end; the distance between the corresponding inner and outer edges of each piston being that which a flat slide valve would have if used under the same conditions. The pistons work in cylinders which are fitted with false seats *X*, called *linings* or *valve chest linings*. The linings are cut with triangular openings for *ports* which lead into a circumferential passage which, in turn, leads into the port or passageways to the ends of the cylinder. A development of the lining with the ports is shown below the valve. The linings are of hard close-grained cast iron; they are accurately turned and faced to fit the casting, bored for the valve, forced into place and secured by screws, tapped half into the liners and half into the casing.

Packing Rings—These rings are shown at *R* in Fig. 36, and are made of hard cast iron. They are turned slightly larger than the bore of the lining, cut obliquely and the ends bolted together by bolts on the inner side. The different forms of rings and method of fitting them are shown in Chapter VII. The ring is held in place by a *follower F* which is secured to the valve by stud bolts and nuts, the bolts being made square where they pass through the follower, with square follower holes, in order to prevent turning and slacking back. The nuts are further secured by split pins or by wiring the nuts.

Live steam may enter through steam pipes at the ends of the valves and exhaust through the center, corresponding to outside valves, or, enter at the center and exhaust from the ends. The valve may be single ported or double ported, the latter is more complicated and owing to the greater port area obtained with the piston valve they are not as necessary as with the flat slide valve.

A second form of the piston valve is used when it is desired to lead the steam pipe to but one end of the valve chest, while the exhaust is led from the central opening or vice versa. This type is

illustrated in Fig. 37, and has open ends connected by a central passage. Steam enters through the opening *A* and is admitted into the steam port through the inner edges of the valves, the exhaust steam

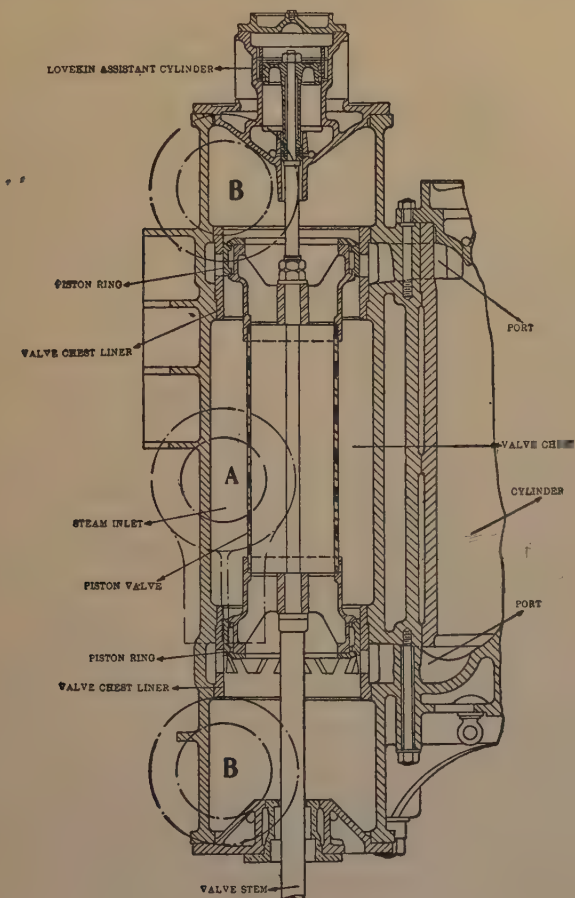


FIG. 37.

from the upper end passes into the upper end of the valve chamber directly to the exhaust *B*, while that from the lower end passes from the port to the lower end and up through the valve to the exhaust pipe.

The piston valve has the advantage of shorter travel and reduced friction, but it possesses the disadvantage of loss from the greater liability to the leakage of steam and to the large clearance space, as the volumes of the steam passages are necessarily greater with this type of valve.

Balance Pistons.—In vertical engines the weight of the valve and its connections comes upon the valve gear and eccentric. To balance this weight and relieve the eccentric, balance pistons working in small cylinders over the valve chest are often fitted, as in Figs. 33 and 36. The lower part of the cylinder is in communication with the valve chest. The area is designed so that this pressure acting upon the under side is sufficient to balance the weight of the valve and connections. The upper end of the cylinder is in connection with the exhaust.

With piston valves this balancing can be effected by varying the diameters of the two ends of the valve, as was done in the design of the Brooklyn. The excess pressure on the large piston balances the weight of the valve and connections.

In recent engines assistant cylinders, described below, are fitted in preference to balance pistons.

Fitting of Valve and Chest.—The valve stem is of steel and is accurately ground and polished where it passes through the stuffing box in the lower cover of the valve chest. In vertical engines, the lower end of the valve rests against the collar on the stem with an adjustable washer between, the upper end being secured by jam nuts and a split pin. The stem is continued upwards, the balance piston being secured to this end by a taper fit and nuts with split pin.

Before the false seat or working linings are put in, the steam and exhaust passages in the valve chest casing are cleaned and *pickled* and then measured to see that the area is no where contracted below that specified. The top and bottom of the chest for a piston valve are closed by removable covers.

Peep-holes are generally fitted near the ends of the valve chest so that the setting of the valve can be observed. Lugs are fitted in the chest to prevent the packing rings or piston valves from overriding the linings when the valves are disconnected from the gear.

Each valve chest is fitted with an adjustable spring **safety valve**, loaded to a pressure suitable to the cylinder; also with a drain cock or valve, operated from the working platform, and discharging either through a trap into the condenser or directly into the bilge.

Lovekin's Assistant Cylinder.—This assistant gear for valves, shown in Fig. 38, is designed not merely to carry the weight of the

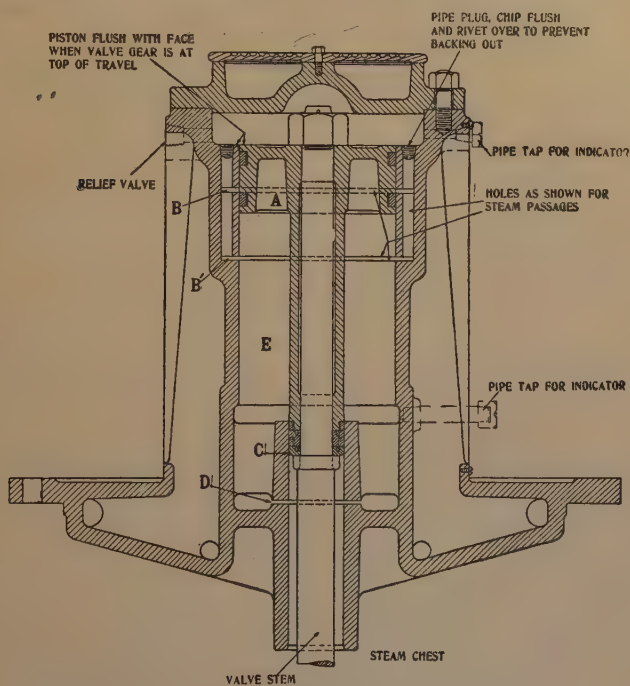


FIG. 38.

valve and gear, but to absorb the momentum and overcome the inertia of these moving parts at each end of the stroke. There is no exhaust port, and the only steam used is that due to condensation. Study of the dimensions of piston *A*, distance between the ports *B* and *B'* and between the packing at *C* and the port *D* will show how *compression* at the proper points overcomes inertia and stores power to start the valve on its return stroke. Steam enters cylinder *E* from the valve chest in the case illustrated; it is sometimes taken from a higher pressure source to reduce the size of

cylinder needed. A great saving in weight is effected by assistant gear permitting the use of smaller parts, and wear on the gear is much reduced.

Setting Slide Valves.—From an analysis of the Zeuner diagram it is seen that the proper distribution of the steam depends upon, (1) the throw of the eccentric, (2) the angle of advance, (3) the steam and exhaust lap, (4) the length of the valve stem. Of these items the only ones capable of variation without making new eccentrics and practically new valves are the length of the valve stem and the angle of advance.

After the complete engine is erected the valves are set for each cylinder separately. Taking one cylinder and its valve, the following is the operation for the ahead motion: Connect all parts of the valve gear, remove valve chest cover, and in the case of piston valve, the peep-hole covers from the shell of the valve chest, in order that the movement of the valve may be measured and noted. Run the link over and secure it for ahead motion. The correct angle of advance having been obtained from the diagram, corresponding key-ways have already been cut in the shaft and eccentrics, so that the latter will take their required position on the shaft. The backing eccentrics are usually secured at once, with their keys. The ahead eccentrics are fitted with adjusting pieces between the key and eccentric in order to effect any slight changes in the angle of advance. If an adjusting piece is not fitted, the key-way in the eccentric may be widened on board and an adjusting piece made.

With all parts connected in place, the crank on one *dead center*, that is, with the center of the crank pin in line with the axis of the cylinder, adjust the valve on the stem by means of the securing nuts and washers, so as to give the designed lead to that end of the cylinder. The shaft is now moved around to the opposite dead center in the direction the engine is to run and the lead measured for that end. It will generally be found that the lead at the opposite ends of the cylinders do not agree with the measurement called for or desired, the position of the valve on the stem must then be altered by an amount equal to the mean of the irregularity at the two ends. The engine is turned to each dead center after each adjustment and measurements taken until finally after as close adjustment of the lead as possible is obtained, the valve is securely fixed to the rod.

As the shaft is moved from one dead center to another the point of cut-off, maximum port opening, etc., are noted for each end.

If after trial it is found that the lead at both ends is too large or too small, this can only be rectified by decreasing or increasing the angular advance of the eccentric on the shaft and not by altering the position of the valve on the stem.

If an angular change of any importance is made in the eccentric's position, the lead should be again examined at both ends and if necessary again adjusted.

After the valve is adjusted for ahead motion, the links should be moved in gear for motion astern and the lead examined at both ends to ascertain if there is a sufficient amount available to enable the engines to back readily.

To Place an Engine on the Dead Center.—In valve setting one of the principal features as already stated is the placing of the engine accurately on the dead centers, and may be done as follows:

Move the shaft so that the crosshead is about two inches from the end of either stroke and scribe a mark *AB*, Fig. 39 across the crosshead and guide for this position. At the same time mark the end of the shaft and its brass, or one of the coupling discs and an adjacent fixed part by the line *CD*. Turn the engine in the same direction as before beyond the center until the marks on the crosshead and slide again coincide. In this position mark on the fixed part the new position *D'* of the mark made on the shaft or coupling. As the axis of the crank pin during this motion has passed through equal distances on either side of the center, by bisecting the angle *DCD'* or taking a point *E*, midway between *D* and *D'* we obtain the position of the line *CE* when the crank is on the dead point.

In this position a center punch mark may be made on the crank and a rigid trammel made to reach some convenient point on the adjacent engine framing or bearing which is also center punched, this latter point when possible is taken so as to be perpendicular to the line of dead centers. The trammel is made of steel rod with one or both ends bent at right angles and both sharpened to a point. The opposite dead center is obtained in the same manner.

The builders usually set marks on the engine frame and turning engine wheel for the main engine centers as follows: Use one point

on the engine frame for all cranks, and with a trammel of fixed length center and as nearly as possible on a tangent, mark the turning wheel, at the same time the crosshead and slide are marked as explained above. On bringing the engine around past the center until the crosshead and slide marks again coincide, mark this second point on the turning wheel with the trammel. Bisect the distance between the two marks; this latter point will give with the trammel the point at which the crank lies on the center. This latter point can be identified by lettering it with the initials of the cyl-

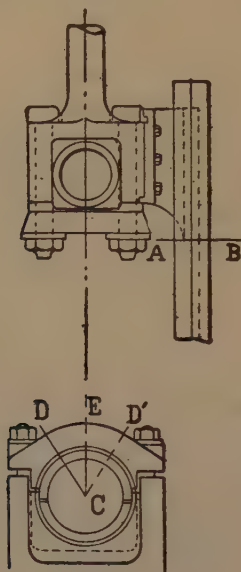


FIG. 39.

inder. The other centers can all be obtained in the same manner on the turning wheel, with the trammel, or when angles of the cranks are known, it then simply becomes a matter of laying off the dead centers of the several cranks from the first one obtained according to the angles of the cranks.

If the turning wheel is not machined, care should be taken that the points marked are at the same distance from the shaft. After a trammel is once made its length should be marked off at a convenient place for reference, by center punch marks enclosed in a square.

CHAPTER V.

VALVE GEAR AND REVERSING ARRANGEMENTS.

Reversing Gear—Link-Motion.—In order that the engine may be worked in opposite directions, suitable reversing gear must be provided, so that the valve may be placed in such a position as to produce revolution of the crank shaft in either direction, and thus drive the ship ahead or astern. There are many types of reversing gear and all usually dependent upon some method of either shifting a single eccentric from its ahead position to another position which will actuate the valve for the reverse direction, or the transfer of the valve connection from one to another eccentric.

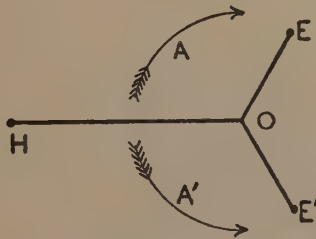


FIG. 40.

To illustrate the first method, the *shifting* or *loose eccentric* may be mentioned as the most primitive form, and one much used formerly in the slow working engines of paddle wheel steamers. Referring to the sketch, Fig. 40, the eccentric must stand in advance of the crank by an angle equal to 90° plus the angle of advance, or an eccentric position OE , with the crank at OH . Under these conditions the engine will run in the direction of the arrow A . To turn the crank shaft in the opposite direction, it is only necessary to bring the eccentric to the position OE' , when for a reverse motion A' , the eccentric will be in its proper angular position ahead of the crank for motion in that direction. This arrangement is

done practically by securing a key or stop on the crank shaft to drive the eccentric, which is loose on the shaft, by pressing against a lug on its side. When going ahead the driving key or stop holds the eccentric in the proper position such as OE . To reverse, the engine is stopped, the eccentric rod temporarily disengaged from the valve stem, and the engine is worked by hand, the eccentric remaining at rest until the key on the shaft comes in contact with the backing lug on the eccentric. The eccentric rod is then dropped into its connection with the valve stem and the engine driven with the eccentric adjusted for backing gear. The arrangement of the loose eccentric is not largely used now, as nearly all modern engines requiring a reversing gear, use either the *link-motion* with two eccentrics, or one of the various forms of *radial gear* with one eccentric which will later be described.

The Link-Motion.—The second method of reversing arrangements mentioned, is the common form now used in modern marine engines, consisting of a combination of links and rods known as the *link-motion*. There are two general types in use; one in which the link is moved on the link block to reverse the engine, known as the Stephenson or shifting link, while in the other, the link is stationary and the block moves in the stationary link to effect the same purpose. Originally designed as a reversing gear, the link-motion can be advantageously employed for the expansive working of steam without the interposition of any other mechanism.

In naval vessels the Stephenson link is the one in general use and will be taken as the type for description.

It was shown in Fig. 40, that with the crank in any position such as OH , the center line of the eccentric for ahead motion is in a direction OE , and for the reversal, in the direction OE' . In the link-motion this is accomplished by having two eccentrics keyed to the shaft, one in position OE and the other in OE' .

Fig. 41 shows the Stephenson link-motion in its simplest form. OA is the crank. OB and OC are the eccentrics keyed in position on the shaft, and connected by the eccentric rods, D and E , to the extremities of a curved bar or link H , whose curvature is that due to a radius equal to the length of the eccentric rod. Within the slot is the link block F over which the link moves, the link block

being connected to the valve stem *J*. *G* is the suspension or hanger rod attached at one end to the saddle plate *L* on the link, and at the other end to the bell crank lever *M*, by means of which the lever can be moved back and forth by the reverse handle *K* and the valve placed in gear with either the forward or backing eccentric. In large engines this movement of the link is effected by a small auxiliary engine.

If the link is moved to its lowest position, the valve stem is nearly in line with the eccentric rod of *B*, and that eccentric controls the action; while on being moved to the highest position the valve stem motion is controlled by *C* and a reverse motion ensues. When in these extreme positions, the link is said to be in *full gear* ahead or astern. By moving the link to an intermediate position, the link block is acted upon by both eccentrics, and the valve receives a mo-

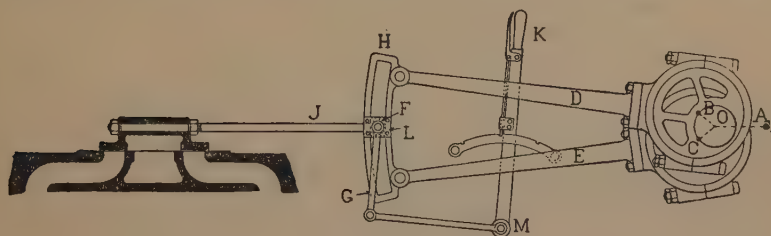


FIG. 41.

tion equivalent to that from an eccentric, of shorter throw and greater angular advance, resulting in a reduced travel of the valve and consequent alteration of lead, cut-off and compression. Hence in addition to furnishing a ready method of reversing an engine, the link is also a convenient means of regulating expansion and steam distribution in the cylinder.

Open and Crossed Rods.—In Fig. 42, if the gear is so connected that when the crank is on the dead center and pointing *away* from the link, the rods are open, the gear is said to have *open rods*; if under these conditions the rods are crossed, the gear is said to have *crossed rods*, assuming that the valve is an *outside valve*. When valves are used taking live steam on the inner edges, or an *inside valve*, the gear is said to have *open rods* when with the crank point-

ing *towards* the link, the rods are open, and said to have *crossed rods* when under this condition the rods are crossed.

To find the length and direction of the equivalent eccentric when the link block is brought into any position between full and mid gear, or as it is technically called, *linking up* in horizontal engines or, *linking in* for vertical engines, connect the eccentric centers D and D' , Fig. 42, by an arc of a circle of radius $= \frac{1}{2} \frac{DT \times DD'}{TN}$.

For open rods this curve should be concave, and for crossed rods, convex to the center O . If the link is linked in to the point M , so that the block is at a distance MT from the point of attachment of the eccentric rod, divide the arc DD' by the point Z in the same

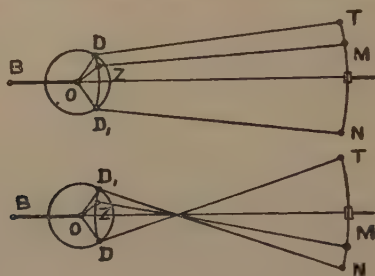


FIG. 42.

ratio that the point M divides the link, OZ then represents the equivalent eccentric, both in length and angular position.

To show the effects of *linking in* by means of the Zeuner diagram, let the diagram be drawn in the usual manner Fig. 43, for the link in full gear, or when the block lies at T , Fig. 44. The full lines in this figure represent the diagram for this condition for one end of the valve, (the lap circle for the other end being omitted to avoid unnecessary confusion in the figure). Admission occurs when the crank is at the position OA , cut-off at OF , release at OR , and compression at OC . On OZ and OZ' as diameters (the length and direction OZ of the equivalent eccentric arm being obtained as above) draw circles cutting the steam and exhaust lap-circles in L, E, G and H . Through OL, OE, OG, OH , draw lines intersecting the eccentric circle in A', F', R', C' , which represent the positions of the

piston at admission, cut-off, release and compression, respectively, at one end of the valve, when linked in to the point M .

It will be seen that the effect of linking in is the same as would be produced by an eccentric whose position on the shaft is at the angle BOZ with the crank, and its eccentricity OZ ; that the admis-

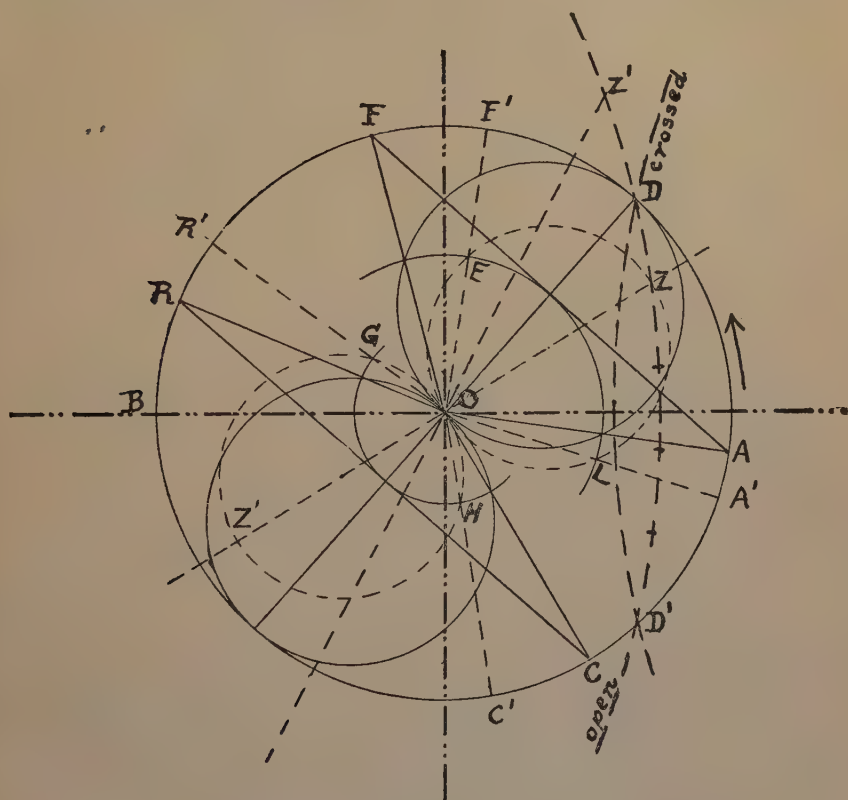


FIG. 43.

sion at A' is earlier, that the magnitude of the lead is increased, that the reduced travel gives a smaller opening to steam and exhaust, that the cut-off at F' , the release at R' and the compression at C' , all occur earlier than when working at full gear.

When the eccentric rods are of the open type, the arc DD' , is

drawn concave with respect to the center O , and Fig. 43, shows the diagram completed for this condition. But, if the eccentric rods are of the crossed type, the arc DD' must be drawn convex to O . Fig. 43, shows the arc drawn in this way for crossed rods but the diagram has not been completed for this condition. By completing the diagram as before, it would be seen that an earlier cut-off is effected by linking in without materially altering the lead, but the travel is very rapidly reduced, giving a corresponding reduction of opening to both steam and exhaust, so that altogether crossed rods are not so convenient as open ones for working expansively by linking in, and are seldom so fitted. Complete the diagram.

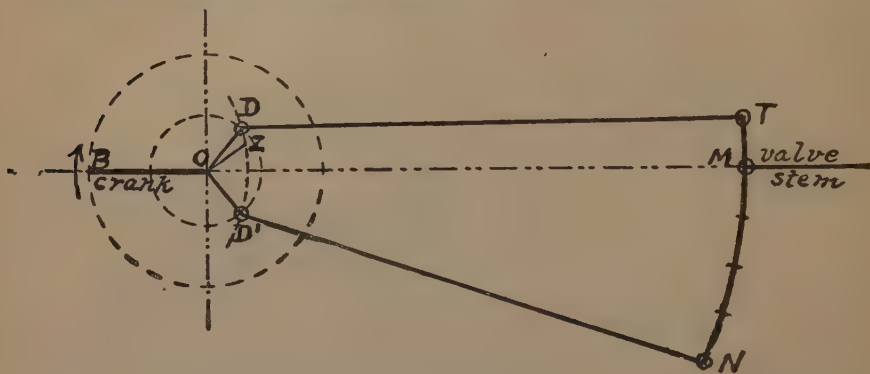


FIG. 44.

Point of Suspension.—Owing to the complex motion of the link, there is always more or less of what is known as *slotting*, or *slip* between the link and the block, and is usually less, the nearer the block is to the point of suspension of the link. In order to reduce the wear and tear consequent upon this slotting motion, the point of suspension is usually taken at the end nearest the ahead eccentric rod. In this case the slotting motion is greatly increased when in backing gear, but in marine work, as ahead motion is the principal one, this disadvantage for backing gear is conceded in order to obtain efficiency when going ahead. When equal efficiency is required for both backing and going ahead, the link is suspended from the center.

Radius of the Link.—Simply as a reversing gear, the curvature given to the link would be immaterial, it might be made straight; but when used as an expansion gear, it is necessary that the valve should move equally on both sides of a fixed point, irrespective of the position of the link. To obtain this sufficiently approximate for all practical purposes, the link is drawn as a circular arc of radius equal to the length of the eccentric rod, that is, the distance between the center of the eccentric sheave and the center of the eccentric rod pin.

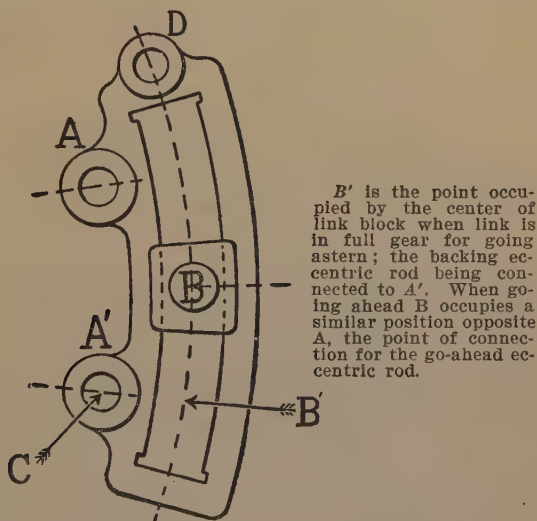


FIG. 45.

Types of Links.—The ordinary form of slotted link is shown in Fig. 45, consisting of a curved bar with a slot in the center in which the link block is fitted. The eccentric rods are connected by pins at A and A' and the valve rod to the link block at B. The point of suspension D is either at the center or at the end nearest the go-ahead eccentric rod. This form of link is very simple, and is much used for small engines. The principal disadvantages are, (1) no adjustment of the link block is possible, consequently any wear causes considerable jar and lost motion in the gear; (2) since the center C of the eccentric rod end does not coincide with the center B' of the link block, the motion given is not very regular.

The *double bar link* is shown in Fig. 46, and is the one in general use both in the navy and mercantile marine. The gear consists of two curved steel bars, *L-L*, secured together at the ends, with distance pieces fitted between the bars. The eccentric rods are connected to the link by means of the pins P_1 and P_2 , two on each side, the eccentric rods having forked ends with adjustable brasses which embrace the pins on each side of the link. The block consists of a steel pin between the bars with top and bottom extensions on each side bearing upon the edges of the bars. Adjustable brass gibs *G* are inserted between the bars and the projections on the link block, and form the rubbing surfaces between the block and bars. The

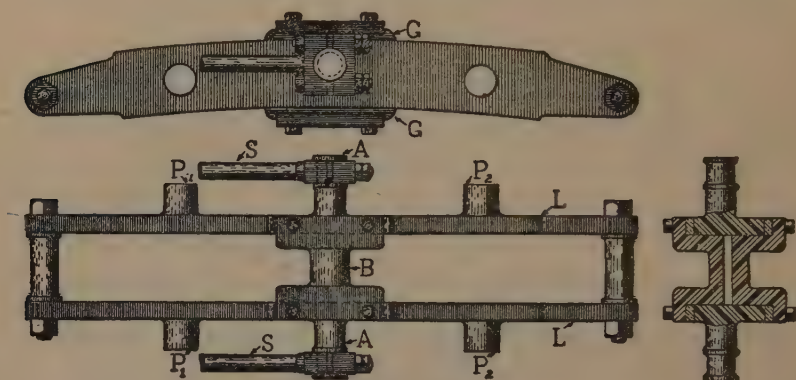


FIG. 46.

valve rod has a connecting rod end with brasses for attachment to the link block pin *B*. The suspension rod *S*, generally two for this type of link, one on each side of the bars, is attached in this case at pins, *A-A*, at the middle of the bars; in other cases, at the ends to prolongations of the eccentric rod pins. All parts can be easily adjusted, and when in full gear the center of the eccentric rod end coincides with that of the link block, so that the motion obtained is very regular.

The principles involved in the two types described are identical, the only difference being in the constructive details.

Independent Linking-up Gear.—In multiple expansion engines, the expansion depends principally upon the proportions of the

cylinders, so that the amount of variation necessary in cutting off can be obtained by *linking in*, and in modern marine engines this is the only means available for increasing the expansion beyond that designed. The links of the different cylinders of a multiple expansion engine are connected to the same rock shaft, so that in linking in, the steam is cut off to the same extent in the various cylinders. It is very desirable to vary the cut-off in each cylinder separately in order to regulate the distribution of power between the cylinders

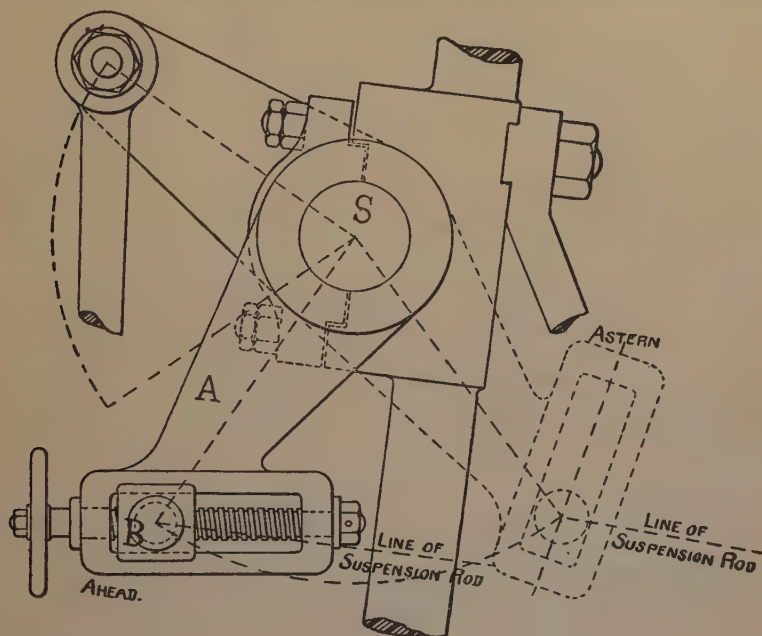


FIG. 47.

and other adjustments. This condition can be effected by means of the following device, shown in Fig. 47:

The reversing arm *A* attached to the rock shaft *S* is fitted with a slot and sliding block *B* to which the link suspension rod is connected and screw gear fitted for moving the block along the slot and thus changing the position of the link to which it is attached. The direction of the slot is so made that when the link is moved over to the astern position, the slot is approximately perpendicular

to the direction of the suspension rod, so that the link is in full gear when in astern position, and if the engines require sudden reversal, the links may be thrown into full backward gear without any readjustment of the independent *linking up* gear. After adjusting the cut-off, the sliding block is clamped in position by means of a nut and washer.

Referring to Plate 5, the general arrangement of the reversing gear is shown as usually applied to the vessels of the U. S. Navy. In the cylinder the steam is regulated by two piston valves whose stems *VV* are rigidly connected to the *valve stem crosshead C* and therefore move as one. *C* moves up and down on the *guide G*, which is bolted to the under side of the cylinder casing. On the lower end of *C* is the bearing for the *link block K*. The *double bar link* is shown at *L* with its *link pins H* and *I*, to which the ahead and backing eccentric rods are secured, respectively, by adjustable brasses as previously explained. The rods are bolted to the *eccentric straps*, the eccentric *A* being shown in full gear. Forged on the middle of the link are the two pins, one on each side, for the bearings of the *suspension rods D*, the other ends of these rods being connected by bearings to adjustable pins *E* on the sliding block of the arm *R* of the *reversing shaft S*. This latter runs along the engine and is fitted with a reversing arm for the link of each cylinder, and is also fitted with arms connected to a small reversing engine, by means of which the links of all cylinders are moved through their blocks and the engine stopped or reversed. The independent linking-in gear with its sliding block for each cylinder is shown at *E* on the end of the arm *R*.

Steam Reversing Engines.—With large engines fitted with heavy reversing gear, it is essential to provide, in addition to the hand or screw reversing gear, a power gear for moving the reversing shaft, and to hold it in complete control at any desired position. Plate 6 shows the reversing engine and gear of the Alabama, together with a skeleton sketch of the link work, and is the general type in use for the propelling engines in the U. S. Navy.

A is the steam cylinder, whose piston rod is attached to the crosshead *B* and continues up to a piston in an oil-controlling cylinder *A'*, the use of which will be explained later. On each

side of the crosshead is attached a link or connecting rod, connected at the other end to the arm *C* of the reversing shaft *D*, so that on steam being admitted to either end of the cylinder, the arm *C* is moved up and down and carries the links to any desired position through their connection to the reversing shaft.

Since it is necessary that the links should be quickly brought to, and held in any position, provision must be made for automatically shutting off steam entering the cylinder *A*, and releasing it as soon as the links have reached the desired position. This is accomplished by means of a series of levers and links in the following manner:

The steam to the cylinder *A* is controlled by a slide valve *V*, usually of the piston type and with little or no lap, so that when covering the steam ports but little motion is needed to open them. The valve stem is prolonged and also forms the stem of the valve of the oil-controlling cylinder previously mentioned. The valve stem is connected by a system of bell crank levers and links to the *reversing lever E*. To illustrate the action, let the gear be in the position shown, move the reversing lever to the right, the link *MN* is raised, and, since *K* is stationary, *J* becomes a fixed center for the moment and the valve stem *I* is raised, admitting steam to the top of the cylinder, since *V* is an inside valve, the piston descends, carrying the reversing shaft arm *C* with it, and the links by their connections are thus brought over. In moving the reversing lever, if it be stopped at any instant, the entire gear, piston, reversing shaft and link will be held in that position, for the point *N* now becomes a *fixed* center, and the piston moving down with the arm *C*, turns *K* upward, and the valve stem downward from the fixed center *N*, thus closing the steam port. This will occur for every position of the reverse lever, so that if the lever is moved but slightly, the piston will move a short distance, when the valve will be brought into mid-position again, and all parts come to rest. If the lever is thrown all the way over, the valve will be opened wide, and the piston will bring the link to the extreme position, and then come to a stop. If motion is started by moving the reverse lever slightly and left to itself, the piston and gear will move along at an equal rate with the lever, and continue until the reversing lever

comes to the limit of its motion. In this case, the friction of moving the reversing lever is less than that of moving the valve, the point *I* becomes for a time the fixed center, until the lever reaches its stop, when *N* becomes the fixed center and the valve is closed.

It will be seen that the gear tends to stop itself, and as it begins to move as soon as *E* moves, it will stop as soon as it has caught up with *E*.

In order that the steam piston may not throw the links over too suddenly, with the risk of breaking some part of the machine, the piston rod of the reverse cylinder is continued and connected to another piston *A'* working in a second cylinder filled with oil. The valve stem is continued up to join the distributing valve *V'* of this cylinder and the ports are so arranged that as the piston moves up and down the oil from one side is forced through small pipes *P* and *P'* to the other side direct, or through an intermediate reservoir. The friction of the oil in these passages thus prevents the steam piston from slamming the link over and checks any violent action. This friction may be increased or decreased at will by throttling these passages with stop valves *L-L'*.

Should any accident or derangement disable the action of the steam cylinder *A*, the hand plunger pump connected at *F* can be used to force oil on either side of the oil piston in *A'* and the engine reversed, but much more slowly.

When reversing by hand it is necessary to work lever *E* in the same manner as when reversing by steam. When the valve is moved upward it allows the pump to draw from the bottom of the control cylinder and discharge into the top of same, and when moved downward to draw from the top and discharge into the bottom.

A full-size working model of this gear, as applied to the Connecticut and Louisiana, can be seen in the model room of the Engineering Building.

Radial Valve Gears consist of various arrangements of gear for reversing the motion of an engine and varying the expansion without the use of eccentrics, or with the employment of but one, and designed to obtain a better distribution than is usually given by the link-motion. Many different forms have been designed,

some producing a constant lead with a large variation of cut-off, without reducing the port opening and slow closing of the exhaust, as in the case of the link-motion. As applied, however, to modern naval practice, the Stephenson link-motion can be designed to accomplish almost as much, and at present in the navy this latter gear is the one in general use.

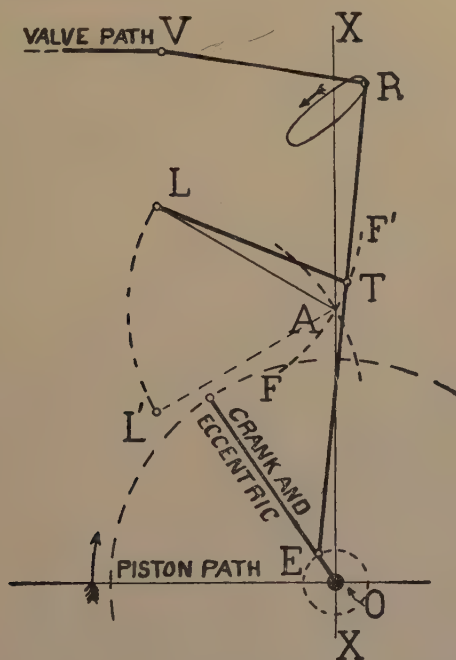


FIG. 48.

Marshall Valve Gear.—This type of the radial gear was fitted to the horizontal engines of the Yorktown class, the Philadelphia and several others, as it does not interfere with the working gear in that type of engine to the same extent as does the link-motion. The gear consists essentially of a rod obtaining its motion at one end from an eccentric on the crank shaft, while the other end is constrained to move in an arc of a circle. Motion is given to the valve by a rod connecting the valve stem to some point on the eccentric rod above mentioned.

The gear as usually applied to naval vessels is shown in **Plate 7**, and the skeleton diagram, **Fig. 48**. OE is the eccentric set with the crank, ER the eccentric rod, LT the radius link attached to the eccentric rod and swinging about the point L on one arm of the bell crank lever, or quadrant QAL , VN and $V'N$ arms on the rock shaft $V'V''$ link connecting valve stem to rock shaft arm. The quadrant QAL swinging on the pin A is used for reversing and changing the cut-off by raising or lowering L , through the rod $Q'Q$ connected to a steam reversing engine. The center of the reversing quadrant is on a line through the center of the shaft and at right angles to the center line of cylinder and coincides exactly with the guided point T when the engine is on either center, consequently the lead is the same for all points of cut-off.

In action, the guided point T of the radius link in oscillating about the point L moves in the arc of a circle FF' , inclined to the line through the center of the shaft. In lowering the pin from full to mid gear, thus reducing the travel and cutting off shorter, the curve approaches the direction of XX and at the mid gear almost coincides with it, and the engine is stopped.

As the pin is still lowered to full backing gear, the inclination of the curve is reversed and steam admitted to the opposite end of the cylinder and the engine reversed.

The eccentric is set at either 180° or 0° with the crank, depending upon the point of connection of the valve stem to the eccentric rod, whether it is between the eccentric and guided point T , or outside of T , and also whether an outside or inside valve is used.

The advantages of the Marshall gear are: constant lead, lightness, compactness, sharp cut-off and good steam distribution; while the objections are: the number of parts in continuous wear, requiring them to be substantially made and adjustable, and the great irregularity of the maximum port opening at the top and bottom ends.

Joy's Valve Gear.—**Figs. 49 and 50** show another type of radial gear in which eccentrics are dispensed with altogether, and the movement of the valve stem taken from some point on the connecting rod. A vibrating link AB has one end attached to the point A on the connecting rod, the other end is connected to the lower end

of a suspension bar or radius rod *C*. The point *A* moves in an ellipse while *B* moves in the arc of a circle. A lever whose long

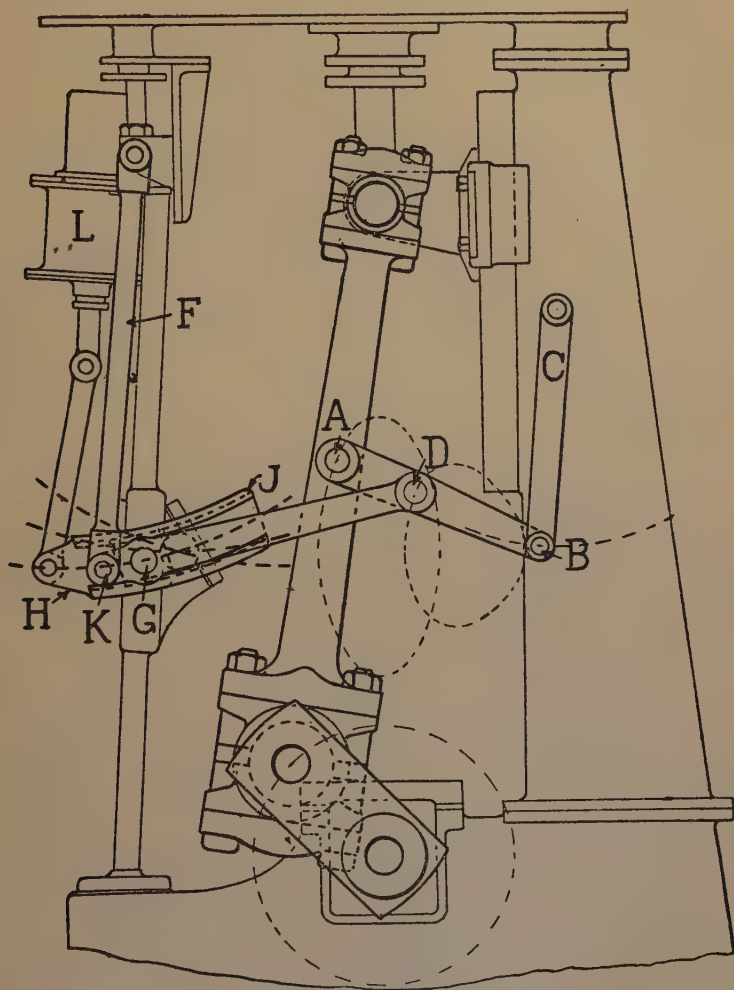


FIG. 49.

arm *GD* is connected to the vibrating link at *D*, while the small arm *GK* works the valve rod, through its connecting rod *F*. The fulcrum of this lever is carried by a pin *G* attached to a sliding

block working in the curved slot *HJ* carried by the reversing shaft.

By the action of the connecting rod at one end and the block working in the curved slot at the other, the extremity of the small arm is constrained to move in an oval curve. In mid gear with the engine on the center this curve is horizontal, and the valve will simply move back and forth on its seat, opening each port alter-

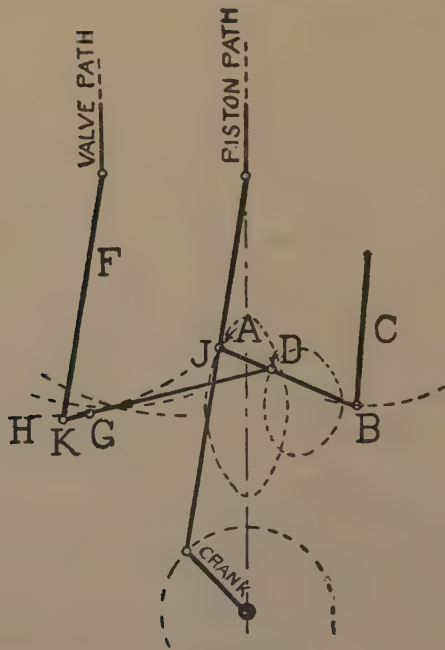


FIG. 50.

nately by the amount of the lead. If the angle of the curved slot be altered by a hand wheel or reversing gear, the curve described by *G* will be of the same form, but will be inclined to the horizontal and the valve travel will be increased. Similarly, if the curved slot be moved to the other side of normal, the curve described will be the same as before, but owing to the altered inclination, the valve will be brought down to the opposite direction and the engine reversed. By varying the inclination of this curved slot from full

to mid gear, any desired range of expansion can be obtained, due to the variation in the travel of the valve. The curved slot is brought over to any desired position by means of screw gear or a reversing engine *L*.

There is much similarity between the Joy and Marshall gear, the point *D* describing the irregular oval in Fig. 49 and acting on the lever *DGK* is equivalent to the center of the single eccentric in the Marshall gear. The motion obtained is almost perfect for a slide valve, the openings at both ends of the valve are equal, and early cut-off can be obtained without excessive leads and compression or premature exhausting. Constant lead for all points of cut-off is secured, for when the engine is on either center the point *G* coincides with the center of the reversing shaft, so that in this position any movement of the curved slot does not affect the position of the slide valve. The objections are, the number of pins liable to wear, and interference with the accessibility of crank pins and crossheads when in operation.

Differential Reversing Valve.—In small engines, such as steering, jacking, anchor and turret engines, when it is undesirable to fit double eccentrics with their connections, though it is necessary to run the engine in both directions, a device shown in Fig. 51 is used.

If the exhaust chamber of a cylinder be made the entrance for live steam, while the steam chest be made an exhaust chamber, the motion of the engine will be reversed. The *differential reversing valve* enables this to be done as desired and furnishes a ready means of reversing the direction of motion.

The engine is fitted with a single eccentric keyed to the shaft at right angles to the crank, and consequently the *distributing valve* is without lap or lead, since it must be capable of starting the engine in either direction. The distributing valve is contained in its own steam chest and is connected to the eccentric in the usual manner.

The *differential valve* is in a separate chamber and to this chamber are connected the separate permanent steam and exhaust pipes *N* and *P*. Two passages *M* and *O* lead from the separate valve chest of the *distributing valve* to the *differential valve chest*,

and it is by making these act as either steam or exhaust pipes that the engine is made to move either way.

If the differential valve is raised, steam entering at *N* will pass into the *distributing valve chest* through *M*, while the exhaust will pass through *O* and out at *P* to the condenser or atmosphere. If the differential valve is made to descend by either a hand lever or wheel, the ports *N* and *O* will be in communication and steam will

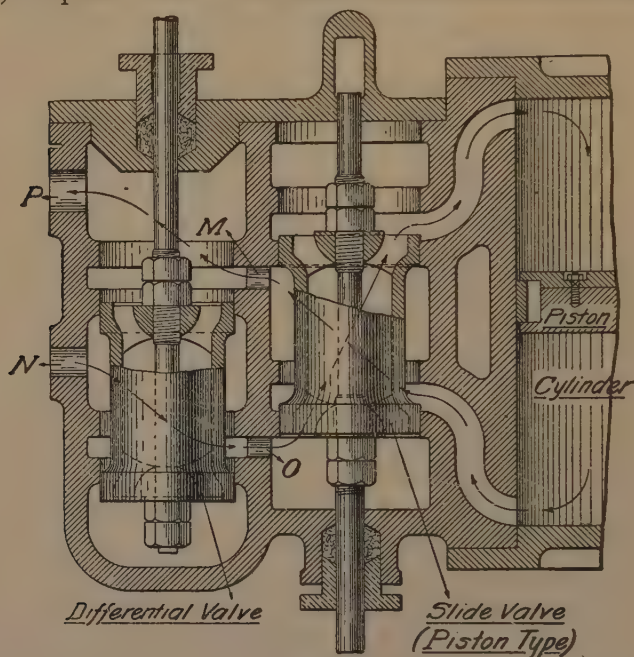


FIG. 51.

enter *O*, through which exhaust was leaving before, and thus the current of steam will be changed and the engine reversed. At the same time the ports *M* and *P* will be in communication and the engine will exhaust through *M* and out at *P*.

Observe that if the slide valve be at its point of mid-travel both ports are closed and no steam can enter cylinder; the engine, if single, would not start until jacked over a little. The same difficulty would exist when the engine is "on a center." There are usually two cylinders with cranks at right angles, so one is always in a favorable position for starting.

CHAPTER VI.

MULTIPLE EXPANSION ENGINES.

Multiple Expansion Engines.—Engines may be classified according to the method of using the steam. In early marine engines steam was admitted to one or more cylinders and expansion took place in the single cylinders or in one stroke, and was then discharged to the condenser. These engines are known as *single expansion engines*. With the increase of steam pressures came the *double expansion* or *compound* engine, in which the cylinders are placed either side by side or tandem, and have the same stroke but different diameters. Steam is directly supplied to the first or high-pressure cylinder where the first expansion takes place. It is then exhausted into the next or larger cylinder, expanding with the increased volume, and at the end of the stroke exhausted into the condenser. The total expansion is the ratio of the volume of the low pressure to that of the high pressure up to the point of cut-off. The triple and quadruple expansion engines are merely progressive stages of the compound engine, a series of cylinders each progressively increasing in size, in which the steam passes from one to the other, doing work in each, and finally sent to the condenser. The typical name given is determined by the number of stages in which the expansion occurs; for instance, in three stages, the *triple expansion*, or, in four stages, the *quadruple expansion* engine, and all known under the general term of *multiple expansion* engines.

The reasons for the substitution of the double expansion for the simple engine will be given, which will also show the causes which led to the replacement of the compound by the triple and quadruple systems as the steam pressure kept increasing.

The total ratio of the expansion of steam in any engine, whether single or multiple expansion = $\frac{\text{Final volume}}{\text{Initial volume}}$, and is approxi-

mately equal to $\frac{\text{Absolute initial pressure}}{\text{Absolute terminal pressure}}$. Thus, if the ratio of the cylinder volumes of a compound engine $= R$, the total ratio of expansion $= r$, and $E =$ ratio of expansion in the H. P. cylinder, then,

$$r = ER.$$

Thus, if the ratio of cylinder volume $= 4$ and the total expansion $= 16$, then the number of times steam is expanded in the H. P. $= \frac{16}{4} = 4$, and the cut-off in H. P. $= \frac{1}{E} = \frac{1}{4}$ stroke.

The principal reasons for the adoption of the compound engine may be stated as follows:

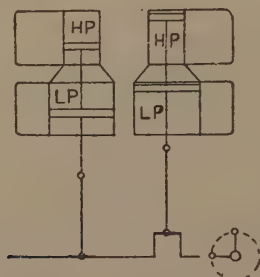


FIG. 52.

1. The maximum stresses are so great in a single cylinder that with high pressures the parts must be abnormally heavy for the power transmitted. In a compound engine, the initial stresses are much less and the ratio of maximum and mean strains much reduced.

2. A greater uniformity of twisting moment on the crank pin and shafting is possible in the compound engine, so that the shafting and framing may be made much lighter and a more uniform and steady motion with better balancing obtained.

3. The amount of expansion required with high pressure steam can be more efficiently and economically carried out in compound engines, owing to the small range of temperature and pressure in each cylinder.

The first two reasons assigned for the superiority of the stage expansion engine are known as the mechanical advantages of the

system and will be explained in detail later, the advantage from the better employment of the heat energy in the steam (3) has been explained in Chapter II.

Up to 90 or 100 pounds of steam the economical advantages are obtained with the compound engine, and beyond this and up to 165 to 185 pounds boiler pressure, triple expansion becomes necessary to obtain the full benefits of the increased pressure and expansion, beyond 185 pounds pressure, quadruple expansion becomes desirable.

There are two principal types of compound engines, known as the "*Woolf*" engine or *continuous expansion type*, and the *Re-*

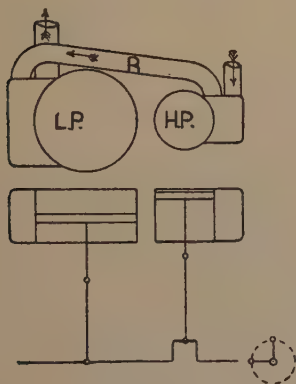


FIG. 53.

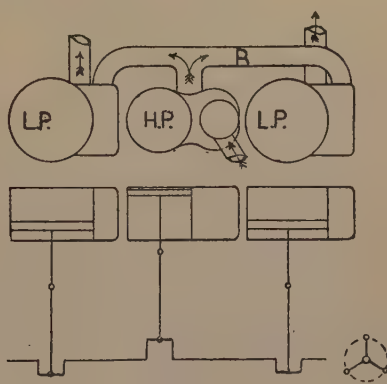


FIG. 54.

ceiver type. In the former the steam is exhausted direct from the high into the low-pressure cylinder, and is only possible when the piston motions of each begin or end together or which act on the same crank, or cranks placed 180° apart. Two forms of this type are much used, the tandem (Fig. 52), either horizontal or vertical, with one cylinder behind the other, the two pistons being on the same rod, prolonged to carry both, and work on the same crank pin. The great source of loss in this type is the large clearance spaces, and the increased length required in the direction of the stroke.

The Receiver type (Fig. 53) consists of the high and low-pressure cylinders placed side by side and each working on cranks

placed at any angle other than 0° or 180° apart, the steam being exhausted into an intermediate *receiver* consisting of the valve chest of low-pressure cylinder and exhaust pipe from the high-pressure cylinder.

As the power increased and the size of the low-pressure cylinder increased beyond the capacity of the machinery space, two low-pressure cylinders (Fig. 54) were used of a combined capacity equal to that of the single low pressure, and the exhaust from the high-pressure cylinder divided between the two low-pressure cylinders. The crank angles for this type were usually 120° , thus giving greater uniformity of turning moment. In a triple expansion engine the exhaust space of the high pressure and the steam chest of the intermediate pressure is termed the *first receiver* and that between the exhaust of the intermediate pressure and the low-pressure valve is called the *second receiver*.

Receivers.—The size of the receiver is largely determined by the design and does not affect the total power of the engine, sufficient volume being allowed for the compression that takes place between the discharge of the exhaust steam into the receiver, and the interval before the succeeding cylinder is ready to take steam. A small receiver increases the back pressure against the high-pressure piston and increases the initial pressure on the low pressure, making the power of the former less and the latter greater, or, as shown on an indicator card, the back pressure line of the high pressure and the admission line of the low-pressure cylinders, instead of being horizontal and parallel to the atmospheric line, are irregular and inclined to the atmospheric line. Similarly, the same reasoning governs the receivers of triple and quadruple expansion engines, generally the necessary volumes in the piping of the connections between the cylinders being found sufficient for the receivers.

Drop in the Receiver is the difference in pressure between the steam at release and the pressure in the receiver into which it is discharged. This is shown on the indicator diagram by a sudden fall at the end of expansion, occasioning a loss in effective pressure, as the steam is allowed to expand into the receiver without doing work. This loss is somewhat reduced by the benefit the steam

receives in becoming superheated, for in unresisted expansion, since heat is given up and no work done, the energy is taken up by the steam and its temperature raised.

The clearances are greater in a multiple-expansion engine than in a simple engine, but this loss in one cylinder goes to fill the clearance in the next, so that the loss is divided.

Effect of Cut-off on Distribution of Work.—In order to obtain nearly equal mean twisting moments on each crank shaft, it is necessary that each cylinder shall contribute the same or nearly the same amount of work, and by varying the cut-offs, the distribution of the total may be adjusted to suit the conditions of the case; at the same time the drop between the cylinders is affected.

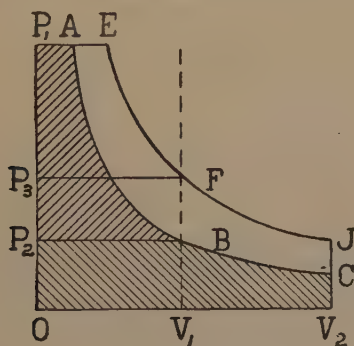


FIG. 55.

The effect of altering the point of cut-off in the high-pressure cylinder with a given steam pressure is to alter the *total power developed* by the engines, since a greater or smaller quantity of steam is used. For instance, in Fig. 55, let OV_2 represent the volume of the L. P. cylinder and OV_1 that of the high pressure. Let the steam be cut off at one-half stroke in the L. P. P_1 = initial pressure; P_2 , the terminal pressure in the high pressure and admission pressure in the low pressure. Assuming no drop, P_1ABP_2 is the H. P. diagram, and P_2BCV_2O the area of the L. P. diagram. Suppose the steam follows one-half stroke in the H. P. while the L. P. remains as before. The H. P. diagram is now P_1EFP_3 , and the L. P. diagram P_3FJV_2O . Hence, the total

power is shown to be greatly increased, and the increased work is nearly all done by the L. P. cylinder, as the H. P. diagram is increased but a proportionately small amount. As the cut-off in the H. P. becomes earlier, it will be found that the larger share of the work is done by the high-pressure cylinder, becoming very marked as the H. P. reaches its minimum cut-off.

Also, altering the amount of cut-off in the high-pressure cylinder alters the *total expansion*; for instance, cutting off *earlier decreases* the power developed and *increases* the *total rate of expansion*.

Unequal distribution of work can be corrected or modified by adjusting the steam distribution in the low-pressure cylinder, cutting off earlier or later. By cutting off earlier the receiver pressure or back pressure in the high-pressure cylinder is increased, hence the work done in that cylinder is decreased. In the low-pressure cylinder the work is increased, for the *total work* must be the same, as it depends only on the initial pressure and total rate of expansion which have not been affected by the change in the low-pressure cut-off. Cutting off shorter on the low pressure has also had the effect of decreasing the loss from *drop*, by making the receiver pressure more closely approach the terminal pressure in the high-pressure cylinder, and with proper values assigned to cylinder ratios, receiver volume, and taking a suitable point of cut-off for the low-pressure cylinder, *drop* can be reduced to a minimum, and at the same time obtain nearly equal distribution of work between the cylinders.

The above can be illustrated by means of a combined set of diagrams of a double or triple-expansion engine. Assume the ratio of cylinders, $1:2\frac{1}{2}:5$, and lay these off on the base line, Fig. 56. Adjust the cut-off on the intermediate cylinder to give a volume up to the point of cut-off equal to the volume of the H. P. cylinder, and in the low-pressure cut-off to give a volume up to the point of cut-off equal to the volume of the intermediate-pressure cylinder. Under these conditions the expansion curve is continuous and equivalent to that obtained from a simple engine whose volume is equal to that of the low-pressure cylinder, and using a volume of steam equal to that in the high pressure up to

the point of cut-off. An inspection of the diagrams will show a very disproportionate distribution of work, the high pressures too small and the low pressures too great.

To obtain greater equality in the power developed in each cylinder, it is necessary to lower both receiver pressures by cutting off later in the intermediate and low-pressure cylinders. Let the cut-off in the intermediate be altered from D to D' (Fig. 57). The admission line of the intermediate cylinder is now given by $C'D'$, OC' , will represent the admission pressure of this cylinder, the terminal pressure of the high-pressure cylinder remaining as be-

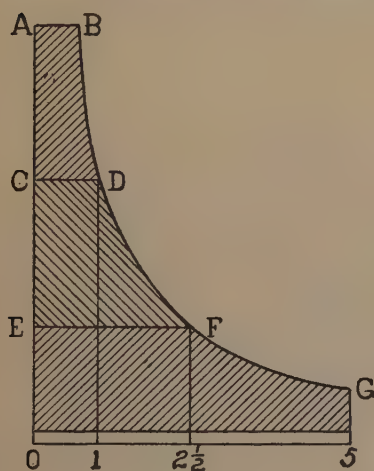


FIG. 56.

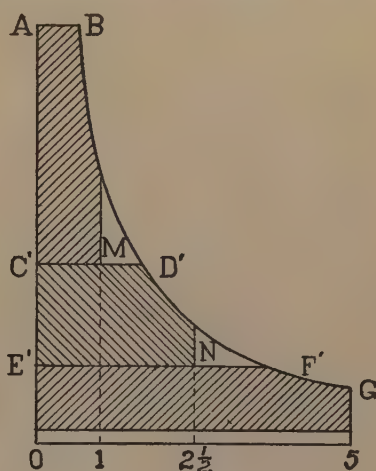


FIG. 57.

fore, and the area of the high-pressure diagram is increased while the intermediate is decreased. A similar action takes place when the low-pressure cut-off is made later, the area of the intermediate diagram becoming greater and that of the low-pressure diagram less, thus dividing the work more evenly between the three cylinders.

Similarly, if the steam is cut off earlier in the intermediate or low-pressure cylinder of a triple-expansion engine, an increased amount of work is done in that cylinder, and a decreased amount in the one preceding, the total amount of work remaining unaltered.

As will be noticed, this alteration of the cut-offs has produced quite a drop between the cylinders, shown by the unshaded portion, and resulting in a loss of work, and which to a certain extent is unavoidable, if equal work is required from each cylinder.

Owing to the proportions that must be adopted in naval vessels, equalization of power, even with the adjustments possible, is not generally secured. At full powers, the high-pressure cylinder develops considerably less and the low-pressure considerably more than its proper proportion.

Separate expansion valves are not fitted to the cylinders, but adjustments within a small range can be made by linking up with the *independent link adjusting gear* described in Chapter V.

Ratio of Cylinders.—In any multiple-expansion engine with a given rate of expansion, the power developed depends entirely upon the size of the L. P. cylinder, and is independent of the size of the other cylinders. Since the L. P. cylinder must contain the whole volume of the steam used per stroke at the terminal pressure, its size is given by that of a single expansion engine of the same power, working at the same initial pressure and total rate of expansion, but a small additional allowance should be made to the piston area for the loss of mean pressure due to friction of passages and *drop*. The quantities given or assumed are the power, initial pressure, ratio of expansion, back pressure and piston speed, from which the mean effective pressure can be determined. Substituting this in the general formula for indicated horse power,

$$\text{Area} = \frac{33,000 \times \text{I. H. P.}}{p_e \times \text{Piston speed}},$$

the area and diameter can be determined.

The ratio between the H. P. and L. P. cylinders is largely a matter of choice; the principal points to be observed are, to divide the power as equally as possible, to avoid excessive drop in the receiver and excessive expansion in any cylinder. That a great deal of latitude and some variety may be expected in the volume ratio of cylinders, can be seen on considering these conditions, which to a great extent are incompatible. It has been shown that by varying the cut-off in the L. P. and I. P. cylinders the work done by the different cylinders may be equalized, and without

regard to the size of the cylinders. But in this adjustment for equal work, the drop will be more or less affected and the initial loads on the piston be greatly varied. Also, in marine work very early cut-offs are avoided, to prevent unnecessarily large initial stresses and to do away with any but the simplest form of slide valve. So under the circumstances a compromise is effected, and in consequence a divergence from uniformity in different designs, but, as stated, in selecting the ratio, the object should be to divide the power evenly and avoid, as far as possible, drop and excessive initial loads. A common rule in general use is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fraction of the stroke completed at cut-off in the high-pressure cylinder. For instance, with a total ratio of expansion of 16 and a cut-off in the H. P. at $\frac{1}{4}$ stroke, the ratio of the volumes of H. P. and L. P. of a compound engine is $16 \times \frac{1}{4} = 4$. The usual ratio obtained in practice for a compound engine using steam of a boiler pressure of 70 pounds is 3.5, and for 90 pounds, 4.0.

Mechanical Advantages of the Multiple-Expansion Engine.—

In addition to the economy in the use of steam noted in Chapter II, the compound or any stage expansion engine, as stated before, possesses *mechanical* advantages over the older type of single expansion engines.

1. The reduction of the range of stress on the working parts, thus enabling a reduction to be made in the size of these parts and giving a lighter engine.

2. A more uniform twisting moment is obtained with increased steadiness and better balancing.

In a simple engine using high pressures and an early cut-off, the difference between the mean and initial thrust is very great, and as the area of the piston is calculated to produce the power with the mean pressure, when the larger initial pressure acts upon it, the initial thrust becomes excessive, so that all parts must be sufficiently heavy to withstand the initial load. In a compound engine the range of pressure is considerably reduced and steam of the maximum pressure comes on a piston much smaller in area, the crank-pin effort is much more uniform, and the shafting and framing may be made much lighter.

To compare the above condition, take the cases of two engines under the same conditions as to steam pressure, ratio of expansion and revolutions, one a two-cylinder single expansion engine, and the other a receiver compound engine with cranks at 90° .

Single Expansion.—Area of each piston equals $\frac{A}{2}$, ratio of expansion equals 6. Initial pressure equals 100 pounds absolute, back pressure, 4 pounds.

$$p_m \text{ in each cylinder} = p_1 \frac{(1 + \log_e r)}{r} = 46.5 \text{ lbs. per sq. in.}$$

$$p_e \text{ in each cylinder} = 46.5 - 4 = 42.5 \text{ lbs. per sq. in.}$$

$$\text{Effective mean load, each piston} = 42.5 \times \frac{A}{2} = 21.25 A \text{ lbs.}$$

$$\text{Effective initial load, each piston} = (100 - 4) \times \frac{A}{2} = 48 A \text{ lbs.}$$

Compound Engines.—Let the ratio of the cylinders be 3:1, and the receiver pressure about 28 lbs. Ratio of expansion in the H. P. cylinder is $\frac{6}{3} = 2$, or the steam is cut off at one-half stroke. To maintain a pressure of 28 pounds in the receiver, since $P_1 V_1 = P V$, $100 \times .5 = 28 \times 3 \times x$, or the point of cut-off in the L. P. cylinder = .59 of the stroke.

$$p_m \text{ in the H. P. cylinder} = 100 \frac{(1 + .6931)}{2} = 84.65 \text{ lbs. per sq. in.}$$

$$p_e \text{ in the H. P. cylinder} = 84.65 - 28 = 56.65 \text{ lbs. per sq. in.}$$

$$\text{Mean effective load H. P.} = 56.65 \times \frac{A}{3} = 18.88 A \text{ lbs.}$$

$$\text{Initial effective load} = (100 - 28) \frac{A}{3} = 24 A \text{ lbs.}$$

$$p_m \text{ in the L. P. cylinder} = 28 \frac{(1 + .5247)}{1.69} = 25.2 \text{ lbs. per sq. in.}$$

$$p_e \text{ in the L. P. cylinder} = 25.2 - 4 = 21.2 \text{ lbs. per sq. in.}$$

$$\text{Mean effective load L. P.} = 21.2 A \text{ lbs.}$$

$$\text{Initial effective load L. P.} = (28 - 4) \times A = 24 A \text{ lbs.}$$

$$\text{Effective mean load H. P. plus L. P.} = 18.88 A + 21.2 A = 40.08 A \text{ lbs.}$$

From the above it can be seen that the initial loads on pistons of the compound engine are equal and less than half that in the

simple engine; the ratio of initial to mean loads in the simple engine is 2.26 to 1, whereas in the high-pressure cylinder of the compound this ratio is 1.27 to 1, and in the low-pressure cylinder 1.13 to 1, hence the compound engine is subject to lighter stresses and to less variation in stress. Consequently the parts may be made lighter and the friction on the guides and journals is much less than with the simple engine.

The curve of twisting moments on the crank shaft is much more uniform in the compound engine, owing to the less variation in the range of pressure and to the lighter reciprocating parts, so that their influence in acceleration and retardation is considerably reduced, consequently a smaller shaft may be used and a more steady motion obtained.

A steam leakage through the piston and valves is more harmful in the single expansion engine, for it then passes directly into the condenser without the performance of work. In the compound engine, leakages past the H. P. may yet do work in the L. P., while a leak in the L. P. is reduced, since there is a smaller difference of pressures on the opposite sides of that piston.

Triple-Expansion Engines.—The same reasons showing the increased efficiencies of the compound engine over the single expansion when the steam pressure was raised from 30 to 90 pounds, can be used to show the superiority of the triple-expansion engine over the compound, as pressures were brought up to 160 pounds, since it is simply an extension of the same principle. It has been shown that there is a decided gain by conducting the expansion in two stages, thus reducing temperature variation; so with higher pressures, increased expansion and the introduction of a third expansion stage, the range of temperature (one of the chief causes of this superior economy) will still be kept within lower limits, for with the same steam pressure, and the same number of expansions, there is the same difference between the initial and final temperatures, and if this is divided among three cylinders instead of two, the range of temperature must be less.

Further, as it has been shown that the compound is superior to the single expansion engine, owing to the decreased range between the initial and mean loads, giving a more uniform turning moment

and a great reduction of maximum stresses on the framework, so a similar reasoning proves, in this respect, the further advantage of the triple-expansion engine over the compound. In other words, the introduction of the triple-expansion engine has been simply an extension of the principles governing the abandonment of the single expansion engine and the adoption of the compound.

As was found with the compound engine, so with the triple expansion, there is no general agreement as to the proportion of the cylinders, the variations in the different designs being generally limited to the intermediate-pressure cylinders. The ratios given are largely governed by the conditions involved. In naval vessels space and weight occupy a prominent position, and the ratio assigned would be largely governed by the adaptability to these two conditions.

As the low-pressure cylinder must contain all the steam of one stroke at the terminal pressure, it is designed as before, of a volume equal to that of the simple engine with the same initial pressure and the same total ratio of expansion. The high pressure, as in the case of a double-expansion engine, is equal to the total expansion multiplied by the cut-off of the high-pressure cylinder. The intermediate cylinder, which is the one in which there is the greatest divergence, is a mean proportional between the high and the low-pressure cylinders, but this as a rule gives results slightly larger than desired and is modified as shown below:

$$\text{Area of H. P. Piston} = \frac{\text{Area of L. P. Piston}}{\text{Cut-off in H. P. Cylinder} \times \text{total rate of expansion.}}$$

$$\text{Area of I. P. Piston} = \frac{\text{Area of L. P. Piston}}{1.1 \times \sqrt{\text{ratio of L. P. to H. P. Cylinder.}}}$$

In general marine practice with steam of 165 pounds absolute for equal work, the ratio of cylinder volumes are about 1:2.5:7, and for 180 pounds, 1:2.75:7.55.

In naval practice, the proportions are somewhat reduced from that given above, a usual ratio for steam of 160 pounds being 1:2½:5½. For full power, the cut-offs are usually set at about

70 per cent of the stroke, and at reduced power can be regulated to cut off at about 50 per cent, so that the expansion is considerably reduced, especially when running with full power. This is due to the fact that in the navy, weight and space occupied are of vital importance on board ship, and by moderating the expansion a saving in both can be effected for the full power developed. Also, for ordinary cruising speeds, the power necessary does not exceed 12 or 15 per cent of the designed power for full speed. Under this wide variation in working powers, if the full cylinder proportions were employed for efficiency at the highest power, the engine would be quite inefficient when working at reduced powers in cruising. With this condition a compromise is effected in the proportions, and a better efficiency obtained at low powers, at some sacrifice of the size and economy when on a full-powered run; but, when this latter condition is required, economy would not be the leading element demanded.

In the merchant marine, on the contrary, the ordinary power regularly used at sea is nearly the full designed power, and boilers and engines are designed to meet that condition with a steam pressure and ratio of expansion suitable to economy at that power. For this reason a smaller degree of economy can be expected from naval engines in comparison with engines designed for a specified duty at or near a given power.

Quadruple-Expansion Engines are a still further extension of the compound or double-expansion system in which the steam is expanded through four successive stages or through four cylinders. This type is used for pressures of 200 pounds and beyond and for expansions as high as 20 and 25. For the pressures and ratio of expansion beyond that assigned to triple-expansion engines, a gain in economy and rotational effort is obtained, similar to that of the triple-expansion engine over the compound. In these engines, the steam passes from the H. P. to the 1st I. P., thence through the 2nd I. P. and L. P. cylinders, and finally exhausts into the condenser. For reasons previously given four stage engines are not used in naval ships now, though they are used in merchant vessels.

With the increased steam pressure in the later ships of the navy, there has come a general increase in the ratio of L. P. to H. P. cylinders, as shown in the following table:

TABLE II.

Year.	Name.	I. H. P.	Pressure.	Type Engine.	H. P.	I. P.	L. P.
1890	Oregon	11,111	160	Trip. Exp. 3 cranks.	1	1.93	4.72
1892	Iowa	12,105	160	"	1	1.98	4.75
1895	Kearsarge	12,318	180	"	1	2.31	5.42
1898	Ohio.....	12,500	200	Trip. Exp. 4 cranks,	1	2.38	6.19
1900	Pennsylvania..	14,000	250	"	1	2.65	7.11
1902	Washington..	23,000	250	"	1	2.72	7.38
1907	Delaware.....	25,000	265	"	1	2.19	7.79
1910	Texas	28,100	265	"	1	2.61	9.06
1911	Oklahoma	24,800	265	"	1	2.84	9.93

For torpedo-boat destroyers the boiler steam pressure carried is 250 pounds, and the ratio of cylinder volumes is about 1:2.5:6.9.

Until within the last few years and with the ordinary pressures carried of 160 pounds, and up to 10,000 I. H. P., the designs of triple-expansion engines in the navy were always of the conventional three-cylinder type, but as the power of the engines increased the design of the L. P. cylinder became so large as to necessitate its division into two separate cylinders and the use of four cranks. In naval vessels, on account of the necessity of keeping the main machinery below the protective deck, the cylinders are arranged in line, as shown in Fig. 2, Plate 8. The Washington and Tennessee class of armored cruisers are equipped in this manner, with two L. P. cylinders each of 74 inches diameter. Had one L. P. cylinder been used, the diameter would have been 105 inches, which, irrespective of the athwartship space occupied, would have been an undesirably large size for the piston speed and speed of rotation expected, viz.: 960 feet and 1507 feet, respectively. With four cranks the twisting forces are more uniform, the vibrating forces reduced, and a better balance obtained. Although the length of the engine room is increased about 15 per cent, it is hardly much more than is needed for a proper installation of auxiliary machinery, while the weight is increased about 10 per

cent. All recent machinery designs for new battleships and large cruisers call for the four-cylinder type, and it may be considered the standard type for all high-powered vessels in the U. S. Navy.

With these high-speed engines on the smaller classes of ships, such as the torpedo-boat destroyers, and the great importance of balancing forces, the third stage cylinder has also been divided as in the above class.

Arrangement of Cylinders.—The different methods of grouping the cylinders of triple and quadruple-expansion engines are illustrated in Plate 8, and marked to show the passage of the steam. Fig. 1 shows the general arrangement adopted in naval vessels for triple-expansion engines with three cylinders and three cranks, the general sequence being, high, intermediate and low pressure, commencing from forward. For a more symmetrical arrangement in the engine room, the low pressure is sometimes placed in the middle.

There are many arrangements of the cylinders and cranks in the four-cylinder engine, with advantages claimed for each system. In many cases the larger cylinders with their heavy pistons are placed inside with the lighter moving parts outside, as in Figs. 4, 5 and 6, and assist in balancing. In Fig. 2, Plate 8, is shown the arrangement in use in our navy for vessels now under construction; the following is the order of cylinders from forward:

Forward L. P. cylinder—H. P.—I. P.—After L. P.

The two after cranks are placed opposite each other and the two forward cranks also opposite, the forward pair being at right angles to the after pair. The sequence of cranks is:

H. P.—I. P.—Forward L. P.—After L. P.

A similar arrangement much used in the British Navy places the cylinders in order from forward, H. P.—I. P.—L. P.—L. P., and gives some variation in the lead of steam pipes and receiver pipes in the engine room, shown in Fig. 3. Another system is to place the two L. P. cranks at right angles (Fig. 3-b) and the forward ones opposite. In another arrangement the plan of the three-cylinder triple-expansion engine is adopted, the two L. P. cranks being set with each other (Fig. 3-c) and the H. P. and I. P. cranks 120° with them. The former plan reduces the

extent of the reciprocating forces and thus the vibrations, while the latter gives a more uniform turning moment and enables the engines to be more readily started.

Fig. 4 shows the arrangement of cylinders designed for the Siberia and Korea of the P. M. S. Line. The cylinder volumes have the ratio of 1:2.04:4.00:8.16 with a steam pressure of 200 pounds and 18,000 I. H. P.

In Fig. 6 is shown the tandem system as applied to quadruple expansion engines in the mercantile marine. The H. P. cylinder is divided into two as in the L. P., making six cylinders with four cranks, the passage of the steam from the boiler to the condenser being indicated by arrows. The arrangement of cranks and other details are made for the explicit object of balancing the engine on the Yarrow-Schlicht-Tweedy system, one of the systems to be described later, designed for the purpose of reducing the vibration forces in high-powered, high-speed engines. The reciprocating weights are adjusted by proper balancing, while the cranks are given the position indicated by calculation, in order to achieve the aim desired. A similar arrangement is applied to four-cylinder quadruple-expansion engines as in Fig. 5.

Experimental Efficiency.—Carefully arranged experiments with actual engines have been made for years by the U. S. Navy Department, the British Admiralty and by the engineering societies of both countries, to test the relative merits of multiple-expansion engines. Without entering into details of the data furnished, which may be obtained from the proceedings of the principal American and English professional societies, it may be stated generally that actual results have shown the double-expansion engine to have about 20 per cent higher efficiency than single-expansion engines, and have shown a superiority of 20 to 25 per cent for the triple-expansion engine of 150 pounds pressure over the double-expansion of 90 pounds. A similar gain in efficiency of the quadruple of 220 pounds over the triple of 150 is shown by general experience, but to a less extent.

Experience at sea with good engines shows a fuel consumption per I. H. P. of about the following quantities:

TABLE III.

Year.	Type of Engine.	Steam pressure in pounds per sq. inch at engines.	Coal consumption per I. H. P.
1870	Simple engine.....	40 to 50	2.8
1880	Double expansion engine.	80 to 100	2.2
1890	Triple expansion engine..	150 to 160	1.7
1900	Triple expansion.....	200	1.6
1900	Quadruple expansion....	250	1.4
1910	Triple expansior.....	265	1.40

Fuel consumption is affected not only by the efficiency of the engine, but also by that of the boiler, and by the quality of the coal and methods of firing. The steam consumption gives a more direct measure of the efficiency of an engine when it can be accurately determined.

The following table gives a close approximation to the steam consumption for various types of engines:

TABLE IV.

Year.	Type of Engine.	Steam pressure in pounds per sq. inch at boilers.	Steam consumption of main engines in lbs. per. I. H. P.
1870	Simple engines.....	40 to 50	23.5
1880	Double expansion engines	90 to 110	18.0
1890	3-Cylinder triple expansion engines.	165 Naval service.	16.5
1905	4-Cylinder triple expansion engines.	180 Merchant service.	15.5
1905	Quadruple expansion engines.	265 Naval service.	15.5
1910	4-Cylinder triple expansion engines.	180 Merchant service.	15.5
1910	Brown-Curtis turbines...	¹ 220	13.0
1912	Parsons turbines.....	² 295	13.5
		³ 180	12.2
		210	12.2

¹ High class merchant service.

² Superheat at engine about 80° F.

³ Superheat at turbine about 40° F.

CHAPTER VII.

CYLINDERS—DETAILS, ATTACHMENTS, ETC.

Engine Cylinders.—Large marine engine cylinders are composed of (1) the *body* or *shell*, consisting of the *outer barrel* with the *lower head*, *ports*, *valve chests* and the upper removable *cylinder cover*, and (2) the *cylinder liner* or *working barrel*.

Steam cylinders are made of the highest grade of cast iron, and from the complicated form of the casting the material should be mixed so that it will run readily in pouring in the mould, to ensure a sound casting. The *outer barrel* or *shell* is usually cast with the lower or crank end head, all ports and passages, steam chests and all lugs and feet for attachment to the *columns* or *engine framing*. The upper *cylinder cover* is cast separately and secured by a flange joint and stud bolts to the cylinder shell. The *cylinder liner* or *lining* is cast separately and forms the rubbing surface on which the piston works. To resist the wear of the packing rings, it is made of very hard close grained iron, the simple nature of the casting permitting the use of this material instead of the softer iron used in the cylinder shell.

In Figs. 1 and 2, Plate 9, are shown vertical and transverse sections of the high pressure and low pressure cylinders used in triple expansion engines of naval vessels, together with the valve or steam chests, liners, ports, etc., from which the details can be observed. *A*, is the cylinder cover; *B, B*, steam ports; *C*, bottom head; *D*, cylinder lining; *E*, cylinder casing; *F, F*, brackets or feet for attachment to the supporting columns; *G*, upper manhole cover; *H*, bottom head manhole cover; *I*, cylinder peep holes; *J*, counter bore in linings; *K, K*, piston valve chest liners; *L, L*, false seat in slide valve chamber; *M*, diagonal bridges across steam ports in piston valve chest; *O*, piston rod stuffing boxes; *R*, balance cylinder; *S*, valve chest covers; *T*, steam pipe to H. P. valve chest; *V, V*, H. P. exhaust; *U, U*, steam to L. P. valve chest; *W*, L. P. exhaust.

The cylinder or liner is accurately bored to a uniform diameter, except at a short distance from each end, where the diameter is increased forming the *counter-bore*, so that at the extreme ends of the stroke the piston travels a short distance beyond the main bore, but not enough to permit the piston rings to slip out. By this means the formation of a shoulder at the extremity of the piston stroke is avoided, thus preventing damage as the limits of the stroke are altered by wear and adjustment of bearings.

In the center of the bottom head there is a hole for the use of a *boring bar*, in boring out the interior surface, and which is afterwards closed by a steam tight cover carrying the piston rod stuffing box and gland. In the larger cylinders, in order to provide for the inspection of the lower end without removing the cylinder cover and piston, a manhole is usually fitted at the bottom of the cylinder, as shown at *H* in Fig. 1, Plate 9.

Cylinder Cover.—The top cover is sometimes made of a single thickness and stiffened with radial ribs, especially in small engines. For structural strength in large cylinders the covers should be cast with double walls as shown in Fig. 1, with connecting ribs between; simply stiffening a single thickness with ribs is insufficient, for as they are placed on the side opposite the pressure, the outer edges of the ribs are subject to extreme tension, and owing to the fact of cast iron possessing a lower power to resist tension compared with its strength to resist compression, a nick or a sudden stress is liable to develop cracks which may spread into the body of the metal. The lower head is formed in the same way, but, as stated before, is cast in one piece with the casing. When formed in this manner, the cover and heads can be used for steam jacketing the ends. Some diversity of opinion exists as to the value of jacketing the ends compared with the complication of steam supply and drains, but, in short stroke cylinders, as used in naval vessels, the ends have nearly as much surface as the walls, and for an effectively jacketed cylinder, the ends as well as the barrel should be jacketed. In all recent designs for battleships and large cruisers in the American Navy, all cylinders have jacketed ends.

The usual shape of the covers in the naval service is conical, to correspond to the conical pistons in general use; they are made of

cast iron or cast steel, principally of the former material in all recent specifications.

There is always a slight distance left between the piston and cylinder cover, termed *clearance*, to provide for any irregularities in the two parts, and to provide for alterations in the extreme piston positions due to the wear of the various connections. The *clearance volume* is the volume between the piston at the end of the stroke and the cylinder head or cover, together with the volume of steam port up to the face of the valve, and is usually measured in percentage of the *piston displacement* (the volume swept through by the piston per stroke).

Cylinder Linings.—These linings are fitted to provide a hard working surface that can be replaced in case of wear, and also to furnish a jacket space between the cylinder casing and working barrel which can be filled with steam entirely surrounding the working barrel, thus maintaining a more uniform temperature in the cylinder, and reducing initial condensation with its enormous loss as shown in Chapter II. In addition, it is also used in warming the engines when raising steam preparatory to getting underway. The linings are of close grained cast iron, as hard as can be properly worked, turned and faced to fit the cylinder casing. They are fitted with a bearing at the middle of their length and also at each end, and are secured by a flange at the bottom end with counter-sunk screws, passing through the flanges and tapped into the cylinder casing. The upper end next to the cylinder cover is left free to allow for expansion, and is kept steam tight by a stuffing box as shown, packed with asbestos or other packing material, and held in place by a gland secured to the *liner only*, so that when the liner expands or contracts at a different rate than the casing, the extremity of the liner is free to move and yet maintains a steam tight joint. Another method is to cut out a groove of a dovetailed cross section, partly in the liner and partly in the barrel, and calking within the recess a ring of soft metal. Other methods in use are composed of a copper ring secured to the top of the liner and the cylinder, the copper ring being made sufficiently pliable to allow for the relative movement due to the contraction or expansion of the liner. Fig. 6, Plate 11, shows the method of securing the liner at

the lower end, and Figs. 6, 7 and 8, Plate 11, show three forms of joints used at the upper end.

The jacket space between the liner and the shell should not be less than from $\frac{3}{4}$ to 1 inch. In triple expansion engines the steam for the jacket is taken from the main steam pipe, and it then usually passes successively through the high pressure, intermediate pressure and low pressure jackets by connecting pipes, fitted with reducing valves and spring safety valves to each jacket. The jackets are drained by a drain pipe from the lowest part of the low pressure jacket to an automatic trap fitted with a by-pass, and thence to the lower part of the feed tank and arranged to allow a free drainage of water of condensation. (See p. 38, first par.)

With this method of casting the jacket separately, the complexity of the casting is much reduced, a hard and sound material is ensured for the working barrel, and the liner when worn may be easily renewed.

Valve Chest Liners—False Seats.—For the same reason that cylinder liners are fitted, the piston valve chests require liners, and these are made of the same material: hard, close grained cast iron. They are accurately turned and faced to fit the casing, forced into place, and secured by screws tapped half in the lining and half in the casing. The ports through the liners have diagonal bridges, as shown in Fig. 2, Plate 9, in order to prevent the valve from catching or jamming, especially when packing rings are used for the piston valves; they are made diagonal to better distribute the wear on the valve rings.

As shown in Fig. 1, Plate 9, low pressure valve chests with flat slide valves are fitted with false seats, made of the same material as cylinder or valve chest liners, and secured to the casing by a number of cheese-headed countersunk bronze tap bolts. By connecting the recesses for the bolt heads with shallow grooves, the lubrication between the surfaces is much improved. In all recent large engines piston valves are used on the L. P. as well as the other cylinders; slide valves are now used only in small engines.

Cylinder Lagging and Clothing.—All surfaces from which there may be a loss of heat from radiation, cylinder walls, valve chests,

cylinder heads, valves, piping, etc., are carefully covered with non-conducting material, such as magnesia or asbestos. This clothing material around the cylinders is kept in place by a walnut lagging or Russia sheet iron, made so as to be readily removable for examination or repairs.

Pistons.—The piston is a disc fitted to work steam tight in the cylinder and receives and transmits the energy of the steam to the other moving parts of the engine. It is composed of three parts, the *piston body*, the *packing rings*, which are pressed out against the sides of the cylinder, and the *follower* or *follower plate* to secure the packing rings in place. The principal considerations governing the design of a piston are, (1) it must be strong enough to withstand the load brought upon it by the steam pressure, and hold the end of the piston rod immovably, (2) the packing around the circumference must maintain a steam tight joint with the cylinder walls so as to prevent the leakage of steam from one side to the other, producing a steam waste and increased back pressure, and (3) it should move in the cylinder with as little friction and wear on the surfaces as possible.

There is a great variety in the form of pistons in use depending upon the size of the cylinder and for the purposes for which the engine is intended. Small pistons are usually made of cast iron and formerly all those of large diameter were of this material, made hollow for lightness and fitted with radial ribs for stiffness. In modern marine engines they are of one thickness, of cast steel, and on account of the superior strength of the metal can be made about 40 per cent lighter and are coned or dish-shaped to increase the stiffness.

Figs. 1 and 2, Plate 12, show the usual arrangement for cast steel and cast iron solid pistons, giving the details of the high pressure and the low-pressure pistons of a naval triple-expansion engine.

When made conical and either *solid* or *cored* the pistons of the different cylinders in a stage expansion engine are made of the same depth. The piston is made slightly smaller than the diameter of the cylinder bore so as not to touch the cylinder walls, the rubbing surface being taken by the rings. *A* is the piston body, *B-B*, are

the packing rings, *C* is the follower or follower plate, *D* the springs and *E* a central boss or body for carrying the piston rod.

Packing Rings and Springs.—The simplest form of packing rings much used in small cylinders are known as *snap rings*, Fig. 4, Plate 12, and are made of cast iron, narrow and turned down to a diameter slightly in excess of the cylinder diameter; a diagonal piece is then cut out of the circumference, and they are sprung open and passed over the piston into grooves cut in the circumferential face. Owing to their elasticity and slightly greater diameter they tend to spring out against the cylinder walls, and as the rings fit accurately into the recesses and are so placed that the joints of the several rings are not in line, they effect a practically steam tight joint. In this case no springs are used and no follower plate is required.

For large marine engines, the preceding plan is objectionable, since the rings cannot be removed and examined without removing the piston, and steam passes where the rings are cut. This has been modified as shown in Fig. 3, Plate 12, by using one or two single rings of tough, close grained cast iron, made on the same principle as the spring ring, but fitted between the *piston flange* and *follower plate*; the ring is turned down about 1 per cent larger in diameter than the cylinder. It is then cut obliquely at *a-b*, as shown in Fig. 2, Plate 11, and to prevent the steam from passing from one side to the other through this cut, a channel *c-d* is cut in the ends across *a-b*, and a brass plate fitted underneath with a tongue piece to fit neatly in the slot *c-d*. The plate is rigidly secured at one end of the cut packing ring, and to the other end by bolts, the holes for the bolts being elongated, so that as the parts wear the rings can spring out. For narrower rings, the same principle of a tongue piece can be used on the side of the rings, as in Fig. 3, Plate 11. The rings are fitted to the piston flange and to each other by hand scraping, then the follower plate is fitted in the same manner to the piston body and opposite side of the packing ring, all parts being made steam tight and the rings free to move laterally. The rings are pressed out by various types of springs; the one in common use in the navy is the ordinary steel coach spring, shown in Figs. 1 and 2, Plate 12. They are inserted in the space between the back of the ring and the body of the piston, and

by their elasticity press the rings out against the cylinder wall. These springs possess the objections that the pressure exerted is unknown and uneven, as it is a difficult matter to set each spring to a uniform tension, and the reaction of the piston from the springs acting upon it interferes to some extent with the piston following the sides of the cylinder freely when it becomes worn.

A second general type of spring and ring is that in which the springs do not react on the piston and depend upon some form of spiral spring, back of the piston ring. One type of such packing, known as Buckley's, is shown in Fig. 1, Plate 11, where a spiral coil of steel wire is bent into a circle and placed in the recess behind the rings. The packing ring is cut in two and beveled on the inside, so that the pressure exerted not only presses the rings against the walls but against the piston flange and follower plate, and prevents the ring from becoming slack between the follower and the piston flange. A slight amount of play at this point rapidly becomes worse, with the continual blow due to the rapid change of motion.

It has been found with ordinary split rings that the steam gets behind the piston rings and with high pressures they are pressed out to such an extent that the cylinder walls are soon cut and worn to an excessive degree. To prevent this action a form of *restraining ring* is used in all recent ships in our navy consisting of the ordinary split rings fitted with lugs as in Fig. 36, page 85, and secured by a bolt and nuts, the clearance of the nut permitting the ring to expand to a certain amount only. On first application it works as an ordinary ring and wears down to a perfect fit, usually at the point where there is but little clearance left for expansion. If leakage occurs at any time, additional clearance can be given by slacking back the nuts to any distance deemed necessary. Other methods in use in the navy are shown in Figs. 3 and 4, Plate 11. In the former the lugs *a* and *b* are cast where the ring is to be cut, faced off, and over these two lugs is placed an elongated steel ring with a slight clearance *c* left for the expansion desired. In Fig. 4 the same end is attained by a strap *a-a* secured on one side of the split ring at *b*, and on the other side of the opening the securing nut fits in an oval hole regulating the expansion or spring necessary.

In both these latter forms, if any leakage occurs past the piston, additional spring or expansion can be given to the rings by scraping the ends of the clearance spaces.

Follower Plate.—This plate is a flat ring made of cast iron, cast or wrought steel and secured in place by steel studs screwed into the piston, the bodies of the studs being usually square, passing through the square holes in the follower. This plate holds the springs and rings in position, and with the cylinder cover off, by removing the follower plate these parts can be examined. In the older engines with cast iron pistons, bolts were used, and as they must be frequently withdrawn, they were not screwed directly into the piston, but into composition plugs inserted into the piston body, thus preventing the bolts from becoming fixed or rusted in place, or if the thread became worn, the plugs could be easily removed and renewed. To prevent breakage care must be taken to prevent the nuts securing the follower from working loose and getting between the piston and cover. For this purpose each nut is secured in place by a brass split pin, or else all wired together, while the studs are made square in their body for the same reason. For horizontal engines, the springs are not continued all around the piston, but for about one-fourth of the circumference of the bottom, a solid block is inserted between the ring and piston to assist in taking the weight off the latter, otherwise, the springs at the bottom would be abnormally compressed and those at the top abnormally slackened.

Piston Rod.—The piston rod is connected to the piston and transmits its motion to the mechanism outside the cylinder. It passes through the bottom of the cylinder by means of a steam tight *stuffing box* and is connected at the further end to the *cross-head*, to be described later. The piston rod is made of wrought steel, passes through the piston and is secured by a nut on the other side. It must fit the piston perfectly tight and not draw when subject to shock, the part within the piston being usually slightly coned for this purpose, and also to enable the rod to be withdrawn with greater ease than a tightly fitting parallel rod. In using a tapered or conical end, it is usual to fit a shoulder at the end, otherwise, the rod becomes so tightly wedged as to prevent removal, and at the same time there is danger of splitting the piston from the

wedging action. To prevent the piston turning, a stop is usually fitted to the rod and piston, and when fitted with a parallel end, a collar is forged on the rod to take the thrust of the piston.

Figs. 1 and 2, Plate 12, show the method of fitting up the piston rod for naval vessels. The end is tapered and has a collar forged on the rod fitting in a corresponding recess in the piston. The rod is secured by a composition cylindrical nut with vertical grooves and set up by a spanner wrench. The nut is secured by a locking ring at *F* while the rod is prevented from turning in the piston by a dowel pin *G* between the piston and rod. Like other cylindrical members,

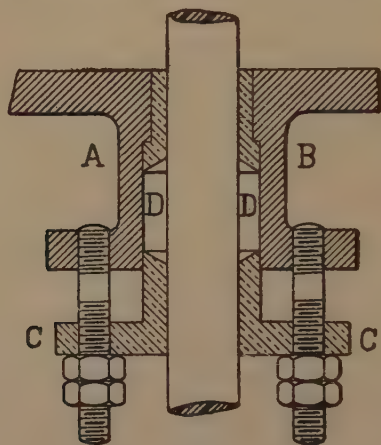


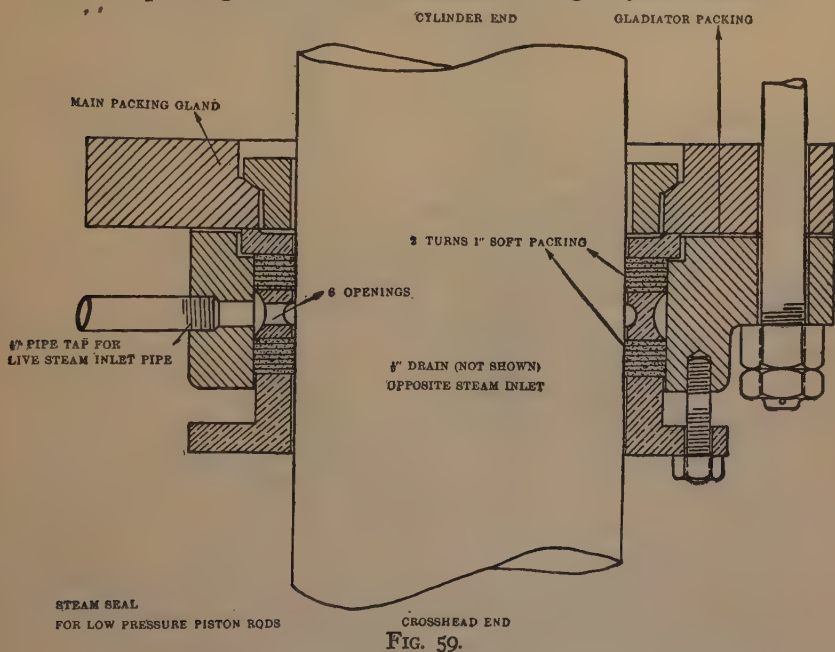
FIG. 58.

such as the shafting, connecting rods, etc., the piston rod is made hollow and closed by a plug at the upper end.

Sketches of other methods of securing the piston rod end are shown in Figs. 9 and 10, Plate 11. On account of the difficulty in withdrawing the rod after a short service, it is not customary to taper the rod end to the full depth of the piston.

Stuffing Boxes.—The piston rod should move freely through the cylinder bottom without leakage, as should all rods entering or leaving chambers which are filled with fluid, such as valve stems, pump rods, pipes where they go through a W. T. bulkhead, expansion joints and all similar fittings. A general type of the

ordinary *stuffing box* is shown in Fig. 58, and consists of a cylindrical chamber *A-B* cast on the cylinder bottom and bored to a larger diameter than the rod. At the bottom of the box a brass bush is fitted, unless the entire box is of brass, and is bored just large enough for the passage of the rod. *C-C* is the gland either of brass or lined with brass, and carried by several studs, by means of which the packing, which is inserted in the annular space, *D-D*, can be compressed around the rod as desired, by the nuts on the stud bolts. The soft packing used in this case is cut in lengths just sufficient to



encircle the rod and placed in the box, each ring of packing breaking joint with the other.

Packing.—There is great variety in the material used for packing, depending upon the fluid used and temperature, and may be generally classified into *fibrous* or *soft packing* and *metallic packing*. The former consists of rubber in combination with some fibrous material as canvas or some form of compound; asbestos, either plain or in combination, hemp, elastic core packings and many other types, cut square, round or triangular section. Packing made of

asbestos fibre has been much used for steam stuffing boxes, but with the higher pressures it is unsatisfactory on account of the rapid wear.

Metallic Packing.—For high pressures this type of packing is essential and is applied to all cylinders of the engine. In this type of packing, as in the other, the greatest variety of detail can be found, and before describing any particular one, it would be well to

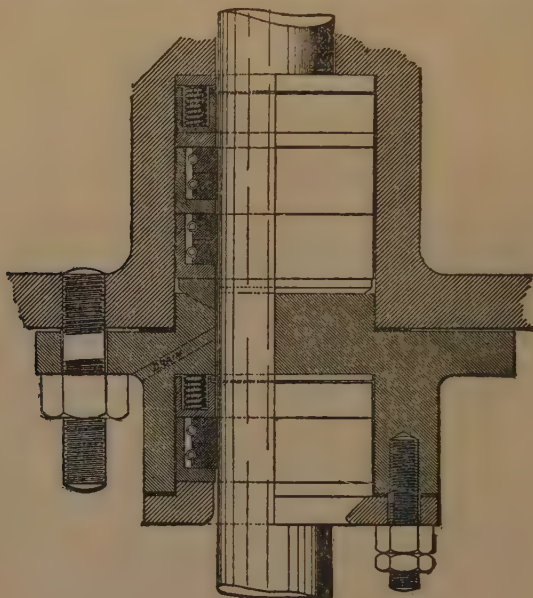


FIG. 59-A.

look at the requisites and purposes of a satisfactory piston rod packing, which may be stated as follows:

1. The packing should prevent steam leakages both along the rod and stem, also between and around the packing, and do this with the minimum pressure and friction.
2. It must be durable under the pressure of the modern high pressure steam, and easily removed and replaced when necessary.
3. The packing should be sufficiently flexible to follow slight lateral movements of the rod, even if the latter is slightly bent or out of line, at the same time maintaining a tight joint between the rod and packing, and between the packing and the stuffing box; and it should not be designed to act as a guide for the rod.

In Figs. 59, 59A, 60 and 61, may be seen the types of packings employed in the U. S. Navy.—Fig. 59 shows a special “steam seal” packing designed to prevent the entrance of air and oil into the L. P. cylinder, where the vacuum existing during part of the stroke would otherwise admit them. The sketch shows how this steam seal may be applied outside any form of packing. A device of this kind is very desirable with the use of forced lubrication, as much oil is thrown upon the piston rods and in the case of the L. P. would be drawn in by the vacuum there existing during part of the stroke, causing much damage to boilers. Excluding air aids in maintaining a good vacuum.

In Fig. 59A is given a sectional view of the Katzenstein improved automatic metallic cup packing, which is used on some of our destroyers and is largely used in merchant marine practice, as well as on stationary engines, with very high pressures. Soft metal gaskets or rings are placed between the gland and the stuffing box, and the gland screwed down tight, making a steam tight joint.

As may be seen in the cut, the packing consists of a series of white metal bearing rings, carried in cups. At the top of the main set of packing, just above the “drain,” are rings with hemispherical joints permitting some vibration of the rod without leaking.

The white metal rings are made in three or four sections each, held in place by helical wire springs. The function of the spring plate is to set the packing rings together. The packing is made of a smaller diameter than the stuffing box to allow for the rod getting out of alignment.

Fig. 60 shows one form of France Marine Type packing. Each of the rings shown is made up of three segments held together by helical springs as shown. The rings are made of soft cast iron for low pressures; for over 200 pounds composition is used for the rings nearest the pressure. At *A* is a gasket making the joint between the flange on the packing case and the face of the stuffing box. At *B* is a connection to which live steam may be led in order to prevent air and lubricating oil from leaking into the cylinder. This steam seal is usually fitted on L. P. cylinders only. The steam is sometimes led into one of the packing ring spaces higher up. *C* is for oil connection.

Another type of piston rod packing shown in Fig. 61, and known as United States packing, has given excellent results and is the

one now used almost universally for ships of the navy under construction during the past few years. It consists of a cone cup (known as vibrating cup), around the rod, into which are fitted soft metal rings. These metal rings are held in the vibrating cup by a spring, and the steam pressure forces them to make a tight

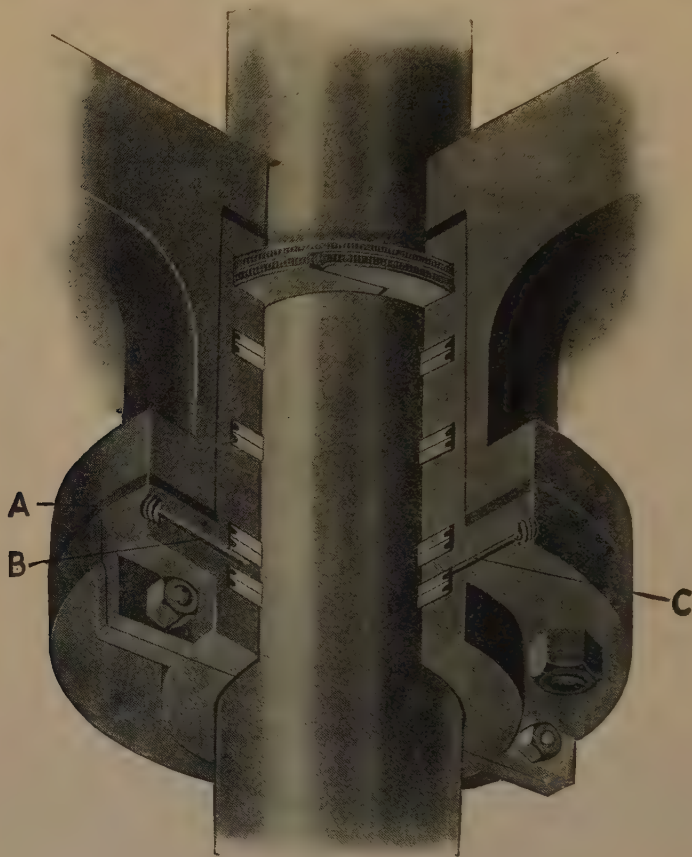


FIG. 60.

fit around the rod, the soft metal, after the joints come together, being forced by spring and steam pressure into the vibrating cup, and hence to the rod by a flow of the metal itself. This forms a steam tight joint and yet is unconstrained laterally, permitting the packing to follow the rod.

The rings which come in contact with the rod and do the actual work of packing are in halves, with a small space between when first applied, so that they may rapidly conform to the shape of the rod and close up. One-sixteenth of an inch is all that is required between ends of packing rings when first applied. Referring to

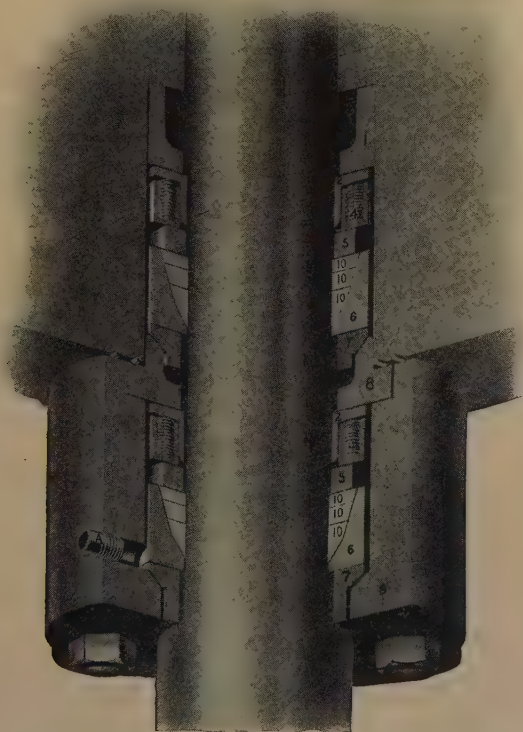


FIG. 61.

the sketch, the following numbers indicate the different parts: 1, preventer; 2, spring guard; 3, spring bushing; 4, spring; 5, follower; 6, vibrating cup; 7, ball ring; 8, ball seat; 9, gland; 10, Babbitt metal rings; A, drain.

Hence the box contains two similar sets of packing elements, top and bottom, and the whole box is secured to the cylinder bottom by stud bolts in the usual manner. The joint between the packing

system and the cylinder is made by a ring of copper wire, shutting off all leaking of steam from the packing space in this direction, while the ground joints and packing ring surfaces prevent it elsewhere.

Cylinder Drain Cocks and Relief Valves.—As water collects in the cylinders and valve chests, either carried over with the entering

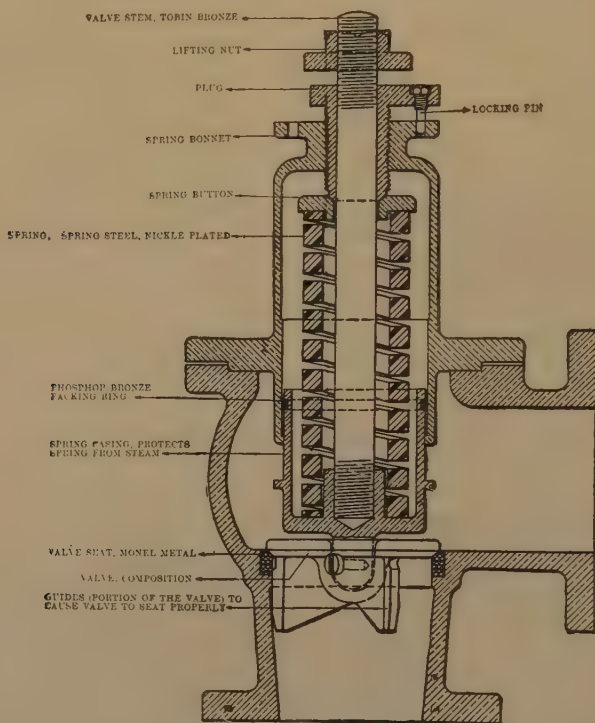


FIG. 62.

steam or from cylinder condensation, especially in warming up the engines in preparing to get underway, it is necessary to provide some ready means of discharging this water. For this purpose, drains are fitted to the cylinders and valve chests and usually consist of small drain cocks placed at the lowest point practicable where water may collect. The drain cock of each cylinder is worked by

a separate lever on the working platform, and all drain cocks discharge into a pipe leading to the fresh water side of the condenser with a branch to the bilge; a spring non-return valve being fitted, which can open to the bilge discharge when the drain to condenser is closed, but preventing the air entering the condenser at any time.

In addition to the *cylinder* and *valve chest drains*, automatic safety or *relief valves*, Fig. 62, must be fitted on each cylinder, top and bottom, so that if water comes over from the boilers in large quantities, as from *foaming* or *priming*, a means of escape will be provided before excessive stresses are brought on the cylinders. For steam of 265 pounds at the high pressure cylinders, the valves are loaded to 275, 130 and 40 pounds, respectively, for the high, intermediate and low pressure cylinders. As a protection against the escape of hot water, the valves are fitted with easily disconnected drain pipes to the bilge.

Indicator Connections.—For the attachment of indicators to the cylinders, there is a cock at each end of each cylinder, as close to the cylinder as possible, and a three way cock for the attachment of indicators on a connecting pipe. For facility of work and for indications as nearly correct as possible, the connections should be accessible and as direct and short as can be fitted.

Receiver Safety Valves and Pressure Gages.—As the low pressure cylinder is constructed for pressures lower than the intermediate, and this in turn for pressures lower than that for which the high pressure is designed, safety valves are fitted on both the intermediate and low pressure valve chests in order to blow off if the proper pressure is exceeded. These valves discharge directly into the engine room and thus give notice of an excess of pressure in these parts. For a pressure of 265 pounds at the engines and with the cylinder proportions in use in the navy the intermediate and low pressure safety valves blow off at 135 and 40 pounds, respectively. *Pressure gages* are connected to each receiver and carried on the gage frame in front of the working platform.

Jacket Fittings.—A steam pipe leads from the main steam pipe to the high pressure cylinder jacket; it is fitted with the *jacket stop valve*. A pipe leads from the bottom of the high pressure

jacket to the top of the intermediate, and a pipe from the bottom of the intermediate to the top of the low pressure jacket. These pipes are fitted with reducing valves, in order that the pressure, and hence the temperature of the jackets may be regulated to that required for the pressure used in the cylinders. On the low pressure side of each reducing valve there is a small adjustable spring safety valve, in order that the designed pressure may not be exceeded; this blows off in the engine room, thus giving warning. The I. P. jacket safety valve is set at 80-130 pounds and the low pressure at 20-60 pounds for naval engines with a working pressure of 265 pounds. Pressure gages are also fitted to each jacket and carried on the general gage board on the working platform.

Jacket Drains lead from the lowest part of the jackets to an automatic trap and thence to the feed tank, in order that the jackets may be kept free of water. A stop valve is also fitted close to the jacket. (See 1st par. p. 38.)

Pressure Reducing and Regulating Valves.—The reducing valves previously mentioned are used for several purposes. They are essential whenever it is economical, necessary or desirable to use a lower steam pressure than that in the boilers, as in most auxiliary engines, in steam jackets, and in the heating system; or where it is advisable to keep a steady lower steam pressure, independent of the fluctuations of the higher pressure in the boilers, as in dynamo engines. Reducing valves are fitted in the steam pipes leading to the auxiliary engines. These valves can be set to any desired delivery pressure, within the limits of the given valve, after which the valve automatically maintains this pressure, regardless of changes in pressure on the supply side.

Two types are in general use on our ships, the Leslie and the Foster reducing valves.

The Foster Pressure Regulator is shown in Figs. 63, 63A and 64. The principle of its action is that of wire-drawing the steam by throttling or restricting the opening at the main valve through which the steam passes from entrance to delivery.

The steam enters valve at *A*, Fig. 63, and flowing in the direction indicated by the arrows, passes out at *B*. In its course it enters chamber *D*, through port *E*, closing valve 4 against the opposing power of the springs 34. Any increased pressure on the dia-

phragms overcomes the resistance of the springs, lifting valve 4 toward its seats. Should the delivery or reduced pressure decrease, the springs overcome the pressure on the diaphragms and force valve 4 open. An equilibrium is thus instantly established. *The desired delivery pressure is controlled by adjusting nuts 36—turn-*

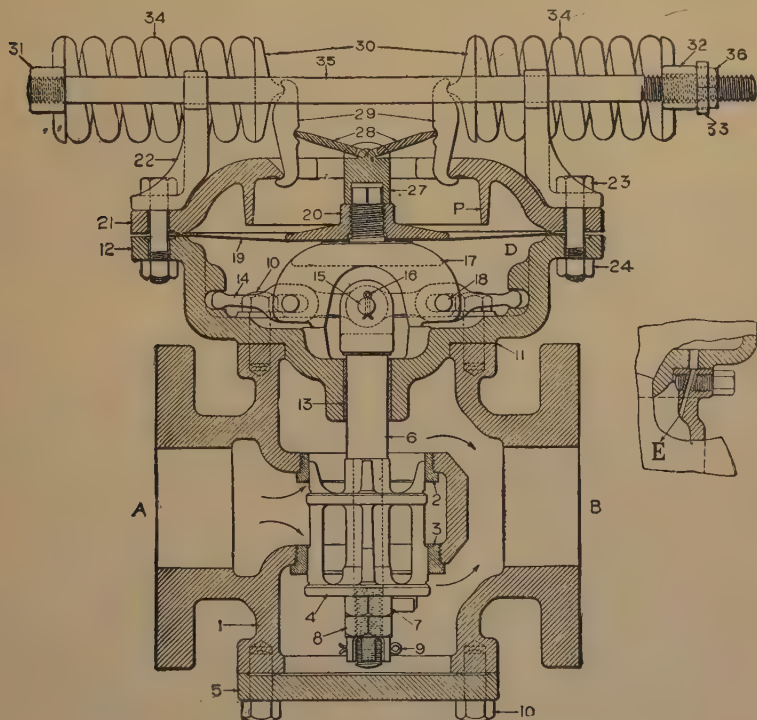


FIG. 63.

FOSTER PRESSURE REGULATOR—CLASS "W."

No.	Name of Part.	No.	Name of Part.	No.	Name of Part.
1	Body.	12	Top.	24	Spring bolt bracket and hood bolt nuts.
2	Upper valve-seat.	13	Top liner.	25	Hood bolts.
3	Lower valve-seat.	14	Valve stem levers.	26	Port screw.
4	Main valve (or clapper).	15	Valve stem pin.	27	Toggle lever base.
5	Bottom flange (or plug).	16	Cotter pin for 15.	28	Toggle lever.
6	Valve (or clapper) stem.	17	Diaphragm center.	29	Toggle lever links.
7	Valve stem nut.	18	Diaphragm center pin.	30	Pivot washer.
8	Valve stem jamb-nut.	19	Diaphragm (set—consisting of one or more).	31	Plain yoke.
9	Valve stem jamb-nut cotter-pin.	20	Diaphragm jamb-nut.	32	Lock yoke.
10	Bottom and top flange screws.	21	Hood.	33	Lock-yoke wrench.
11	Gasket.	22	Spring bolt bracket.	34	Springs (two to a set).
		23	Spring bolt bracket bolts.	35	Spring bolts.
				36	Spring bolt nuts.
				37	Diaphragm chamber plug.

ing to the right increases and to the left decreases the delivery pressure.

The Foster "G" will work equally well in *any position*. One adjusting spring only is used. The valve has no outside parts and is entirely self-contained. Other important features are its removable piston cylinder and the interchangeability of internal parts.

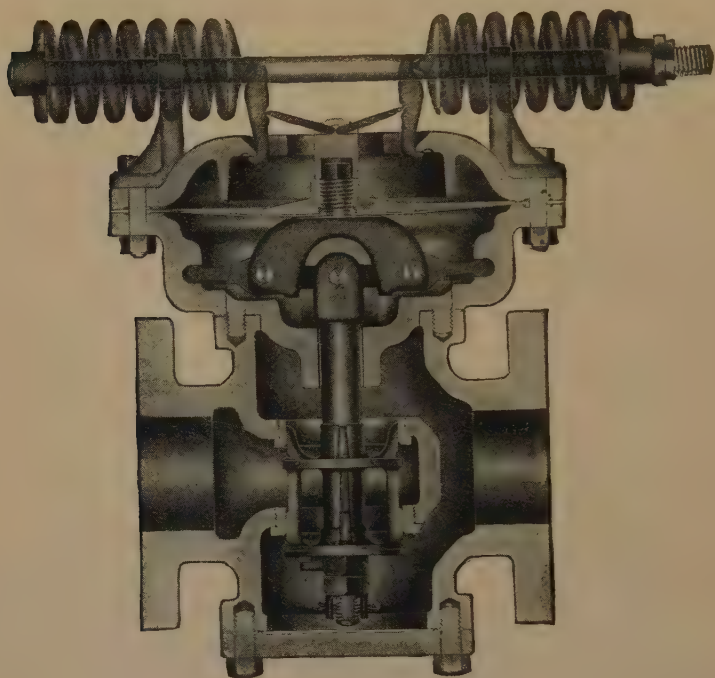


FIG. 63-A.

FOSTER PRESSURE REGULATOR—CLASS "W."

The auxiliary valve internal parts in sizes $\frac{1}{2}$ " to 4" inclusive, are identical and interchangeable, and all sizes from $4\frac{1}{2}$ " to 12" are also identical and interchangeable, thus reducing the necessity for carrying a large assortment of these parts, an advantage in naval service. The single adjusting spring and single diaphragm with which it is fitted, permit a wide range of pressures on delivery side. The slight movement of the auxiliary valve (the controlling

feature) together with its quick response under changeable conditions, makes it a valuable valve for intermittent service, where the load is liable to sudden changes.

Steam entering at *A*, Fig. 64, passes through main valve port *H* to outlet *B*. The initial pressure passing through port *C* enters chamber *P*, thence to the top of piston 2, through ports *L*, thereby open-

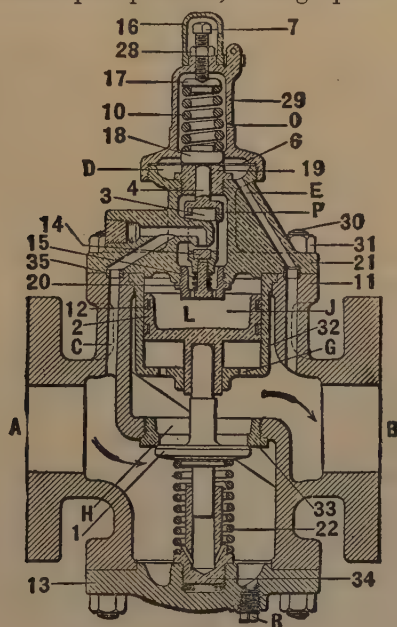


FIG. 64.

FOSTER PRESSURE REGULATOR—CLASS "G."
2" size and smaller made of composition.

No.	Name of Part.	No.	Name of Part.	No.	Name of Part.
1	Main valve.	14	Auxiliary valve disc.	28	Lock nut.
2	Piston.	15	Auxiliary valve spring.	29	Spring chamber.
3	Yoke.	16	Locking device cap.	30	Body studs.
4	Yoke guide stem.	17	Upper spring washer.	31	Body stud nuts.
6	Diaphragm.	18	Lower spring washer.	32	Piston cylinder.
7	Spring adjusting screw.	19	Upper auxiliary valve plug.	33	Seat ring.
10	Adjusting spring.	20	Lower auxiliary valve plug.	34	Bottom flange bushing.
11	Body.	21	Auxiliary valve top.	35	Port bushings.
12	Piston rings.	22	Main valve spring.		
13	Bottom flange.				
A	Inlet.	G	Equalizing ports.	P	Auxiliary valve chamber.
B	Outlet.	H	Main valve opening.	R	Drain (should be piped to trap).
C	Controlling port.	J	Piston chamber.		
D	Diaphragm chamber.	L	Ports from auxiliary to piston chamber.		
E	Port to diaphragm chamber.				

ing the main valve 1. The delivery pressure, passing through port *E*, raises the diaphragm 6 against the pressure spring 10, and allows the spring 15 to close the auxiliary valve 14. The pressure in chamber *J* is then equalized by the reduced pressure passing through ports *G* to the underside of piston 2, and thus allows the spring 22 to close the main valve, which is then held to its seat by the initial pressure.

Upon any reduction of the delivery pressure acting on the diaphragm the spring 10 forces it down and opens the auxiliary valve 14, admitting steam to top of piston 2 as before explained.

The delivery pressure is adjusted by screw 7; tightening the tension of spring 10 increases the discharge pressure, and vice versa. The adjustment once made, the delivery pressure will remain constant regardless of any variable volume of discharge or variation of the initial pressure, so long as the latter is in excess of the delivery pressure.

Turning to the *right* increases and to the *left* decreases the *delivery pressure*.

Leslie Regulating Valve.—Fig. 65 shows a sectional view of the Leslie Valve, which consists of a main body *A*, containing the main valve *D*, main valve spring *E*, piston *F*, diaphragm *G*, controlling valve *J*, with its spring *K*, the adjusting spring *L*, adjusting cap *O*, *R* steam inlet, *T* steam outlet, *S* inlet port to controlling valve and *U* port to diaphragm chamber. The following is an abridged description of the action of this valve, from Bieg's Naval Boilers:

The Action of the Leslie Valve.—Steam enters at *R* under boiler pressure and leaves at *T* under the reduced pressure. The figure shows the main valve closed, as it is when no steam is on *R*. Main valve *D* is held against its seat by the spring *E*. Attached to the upper end of its stem is the piston *F* which works in a cylinder, the top end of which is always in communication with *R* when the controlling valve *J* opens. This valve is held against its seat by the small spring *K*, and its stem is independent of the small diaphragm *G*. The under side of *G* is always in communication with outlet *T*, and its upper side subject to the tension of the adjusting spring *L* acting on the block or seat *N*.

When the valve is to be regulated after being secured in place, the cap *C* is unscrewed until there is no tension on spring *L*. The

drain cocks on the delivery side and at the bottom of the lower bonnet *C* (not shown) must be opened, and then the by-pass valves opened if fitted, or the stop valve on the inlet side opened very slowly until it is opened full. The cap *O* is now screwed down slowly, a little at a time, to allow the regulator to become thor-

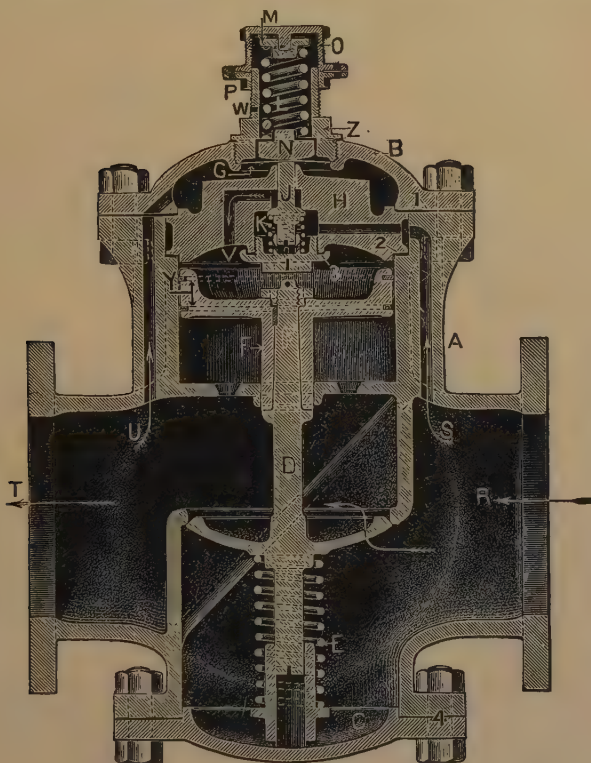


FIG. 65.

oughly heated and drained. When the desired pressure on the delivery side has been obtained (the drains being closed when all water has been blown out and the delivery pressure is steady), the lock nut *P* is set up tight against *O*, and the two locked together.

When the cap *O* is screwed down, it puts *L* under tension. The latter forces *G* down and partially opens valve *J* against the spring *K* and the inlet steam pressure, which has been admitted to the

back of J through the port S . The steam reaches the top of F through the port V and forces the piston and the main valve D down and partially open. Steam is now wire-drawn from R into T and, at the reduced pressure, acts on the bottom of diaphragm G , through the port U , against the tension of the spring L , and acts also on the under side of the piston F , through a number of holes in the bottom of the cylinder. So long as the pressure in T , and, therefore, that under the diaphragm, balances the tension of the spring L , there will be no change in the position of J as fixed by the setting of spring L , and, hence, no change in F and D . But, if the delivery pressure falls, spring L acts downwards and opens J wider, which results in a greater pressure on top of F and a correspondingly greater opening of D , and, therefore, in a greater pressure in T until equilibrium is again restored between the two sides of the diaphragm. The action is similar, only in a reversed manner, when the pressure in T rises.

These valves are of the same construction for all sizes from $\frac{1}{2}$ to 20 inches, except that all valves larger than 4 inches must be fitted with by-pass valves so that the pressure on the delivery side can be raised before the stop valve on the inlet side is fully opened in order to prevent pounding or other injury. The size of the diaphragm does not change with each size of the valve, there being only two sizes used, one for valves from $\frac{1}{2}$ to $1\frac{1}{2}$ inches, and the other for valves 2 inches and larger. This is made possible because the movement of the diaphragm affects only a smaller controlling valve J , and not the main valve D directly.

By-Pass Valve to the Receivers.—For the purpose of assisting in starting the main engines, branch pipes are led from the main steam pipe direct to the intermediate and low pressure receivers by means of which steam can be admitted to the receivers and the warming up of the engine facilitated. Formerly the by-passes were sometimes opened when frequent stopping or reversing was expected; they are not necessary for this and should not ordinarily be used.

Separators.—Separators are designed to be placed in steam lines to remove all water from the steam before it enters the engine; they are described in the text book on boilers. There has been a tendency to omit them entirely, but it may be accepted as correct principle

that they should be installed in the main steam pipe to reciprocating engines. When properly handled they will prevent water entering the steam chest unless it is coming over in large quantities. In main turbine installations they are hardly necessary, since carrying over of water ("priming") in quantities controllable by a separator does no harm to turbine machinery, merely causing it to slow down temporarily. However, in foreign practice separators are always installed, even with turbines. Furnishing wet steam continually to a turbine injures the blading by erosion.

It is claimed by some engineers that separators are ineffective in main steam lines; that they will not prevent water entering the engines if the boilers prime badly; that, if not watched, they even accumulate water and then send it over in a gulp; and that proper design and handling of boilers, *i. e.*, prevention of foaming, is the only real preventive for water in the engines. The same reasoning would apply to dynamo and other important auxiliary steam lines. If the boilers do prime, however, a separator, if properly tended, will be of assistance.

CHAPTER VIII.

CROSSHEAD, CONNECTING ROD, CRANK SHAFT, ETC.

Before discussing the details of the crosshead, the nature of the forces acting at the crosshead end of the piston rod should be understood, and the means taken for their proper control. The mechanism for converting the reciprocating motion of the piston into the rotary motion of the shaft consists of the *connecting rod* and *crank*. As the crank revolves, the angularity of the connecting rod is zero at the commencement of the stroke and increases until the crank stands at right angles to the direction of the piston's stroke and decreases until at the end of the stroke, when it again becomes zero.

Referring to Fig. 67, the piston rod end and crosshead are constrained to move in a straight line by means of the guide XY , while the crank pin moves in the circumference of a circle, the crosshead and crank pin being connected by a suitable link or connecting rod. If the piston with a force P moves towards the crank, and the latter turning in the direction of the arrow, the resistance opposed to the rotation of the crank exerts a compressive force Q on the connecting rod, which may be resolved into two forces, one tending to compress the piston rod, and the other tending to bend it at right angles and producing a downward pressure or pressure to the left upon the guides, which is balanced by an equal reaction R . On the return stroke, Fig. 66, taking the position of the crank at C' , both piston and connecting rod are now in tension and there is again a downward resultant thrust upon the guide. Hence, for this direction of rotation, the pressure is always *downward* or to the left. If the engines are made to rotate in the opposite direction, there will be a complete reversal of all the above conditions, and the thrust now on the guides would be *upward* or to the right, and, consequently, another guide is required on the opposite side of the crosshead to prevent deflection of the

piston rod when working in astern motion. Where engines are required to work as much in one direction as in the other, the areas of contact surfaces between the crosshead and either the ahead or backing guides are the same. For engines required to work principally in one direction, as is the case with marine engines, the surface in contact with the ahead guide is made sufficiently large, while that on the backing guide may be much reduced.

Also, when the crank drags the piston, as when the steam is shut off and the shaft still continues to revolve, the direction of the thrust in the guides is reversed, so that two guides are necessary under all conditions.

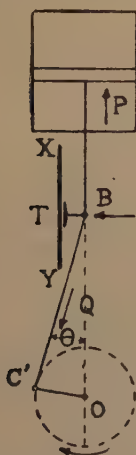


FIG. 66.

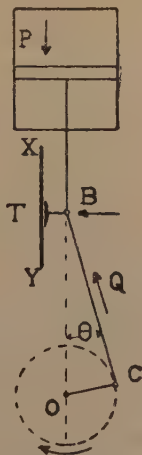


FIG. 67.

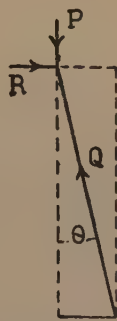


FIG. 67a.

The magnitude of the thrust depends upon the angularity of the connecting rod; that is, upon the angle between the connecting rod and the line of motion of the piston rod, and the thrust may be shown by applying the principle of the parallelogram of forces at the point B .

For any angle θ we have:

$$P = Q \cos \theta \quad (1), \text{ or } Q = P \sec \theta.$$

$$R = Q \sin \theta \quad (2)$$

$$\text{From (1) and (2), } R = P \tan \theta \quad (3)$$

$$\text{Also, } Q^2 = P^2 + R^2.$$

When the engine is on the dead points, $\theta = 0$, and the pressure upon the guides becomes zero, the entire pressure exerted by the piston rod being transferred directly to the connecting rod. The guide pressure becomes a maximum when tangent θ is a maximum, and θ has its largest value when the crank makes an angle of 90° with the axis of the piston rod, hence, the greatest wear on the guide will be found about the middle of its length. The size of the angle θ also depends upon the ratio of the length of crank to connecting rod, the lower this ratio the greater will become the value of θ , other things being equal. For the ordinary ratios of crank to connecting rod, the value of θ is so small that there is but little difference between the sine and tangent, so that the former may be substituted for the latter, in equation (3), and for a practical rule we may write,

$$\text{Maximum pressure on guides} = \text{load on piston} \times \frac{\text{length of crank.}}{\text{length of connecting rod.}}$$

In the design of the surface of the crosshead, the maximum thrust exerted is the one considered, as the guide must be wide enough to carry the maximum load with safety. In modern marine engines the area of the sliding surface of the crosshead is calculated so that the pressure per square inch shall not exceed 80 to 100 pounds per square inch when the guides are made of cast iron.

The Crosshead.—The crosshead is the head or block at the outer or lower end of the piston rod to which the connecting rod is attached, and guides the piston rod in a line through the axis of the cylinder, against the bending moments just described. The crosshead travels between two fixed bars parallel to the axis of the cylinder, known as the *crosshead guides*.

The design of the crosshead varies greatly and is governed by the type of the engine, and may be divided into two general classes, known as the *closed type* and the *open type* of guide bearings. Fig. 68 represents the open type and consists of a body or block of forged steel *A*, with the piston rod connected as shown, and on either side longitudinally are the *crosshead pins BE*, for the

attachment of the forked end of the connecting rod, by means of adjustable brasses. On the remaining two sides are the removable slippers *DD*, whose sliding surfaces are faced with white metal fitted in dovetailed recesses. The slippers slide on the guides bolted to the inverted "Y" columns of the engine framing, shown in Fig. 75, and containing a guide surface on the back as well as on the front side of the engine. The ahead guide is made hollow to permit of circulation of sea water to assist in keeping it cool, and the guide surface is usually scored with channels for the distribution of oil. The crosshead pins, like other parts of shafts and pins, with the high-grade material used and careful design, are fitted with axial holes in them as shown in the figure, and since

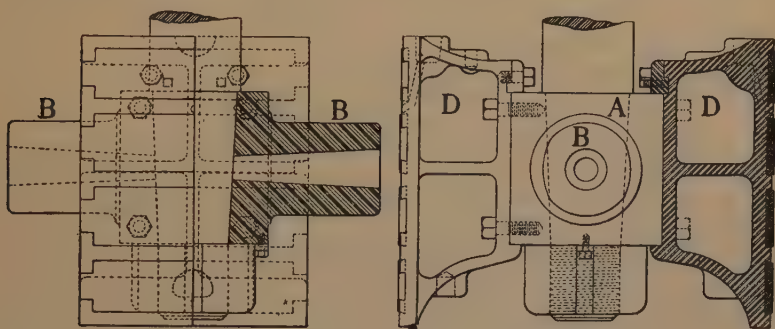


FIG. 68.

the bearings on the journal have an oscillating motion, the pins are flattened on each side, the flats stopping a little short of the bearing surface to prevent the escape of oil.

Closed Guide Bearings are used in the type of engine framing in Fig. 2, Plate 19, where cylindrical columns are used in front of the engine. This type of crosshead is illustrated in Figs. 69 and 70, and is similar to the open type, except that but one wing carries the sliding surface. The ahead guide, as before, is bolted to the engine frame, the backing guides *AA* are lugs of cast steel of the shape shown, and are bolted to flanges provided on the ahead guides. When in motion ahead, the slipper, faced as before with white metal, bears against the face of the ahead guide. On the

reverse motion, the back of the sliding block, which is also babbitted to the proper width, bears against the lugs *AA*. In the figure, *BB* is the ahead face of the slipper, *CC* the backing face, and *D* the arched back. In both forms of guides, adjustments can be made to take up wear by means of liners or packing pieces back of the slippers for the ahead motion, and between the backing guides and the flange to which they are bolted for wear on the backing side.

White Metal.—The various bearings, such as crank pin, crank shaft, crosshead slippers and many others, are what are known as *babbitted bearings* to reduce the friction. The bearings to be babbitted are cast with a number of shallow pockets on the surface,

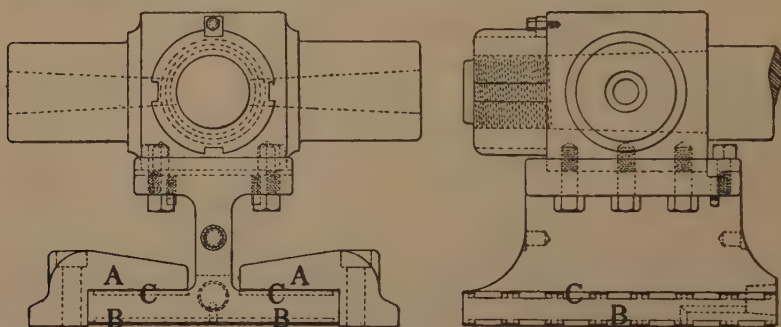


FIG. 69.

of various forms, square, oval, round and sometimes grooves similar to the *lignum vitae* grooves in stern bearings. The edges of the pockets are raised slightly above the surface so as to retain the metal in place. The white metal is then melted and run into the pockets, and after cooling it is well hammered or peneled in place, in order to compress and harden the metal, and fill up all crevices within the pockets. The white metal bearing is then machined down to the proper dimensions and hand-scraped to a bearing surface, the channels between the sections of white metal serving to supply oil to the different parts of the bearing surface. White metal is an alloy of tin, copper and antimony, and forms a soft bearing surface, working with but little friction and under ordinary temperatures is firm enough to sustain the highest pressures on

the rubbing surfaces if properly supported by the composition backing. If the bearings should become hot the white metal will soften and run.

Spare brasses are carried for each part, and the defective ones can be refitted with white metal, or put in working shape on board. There are various forms of white metal in the market, known as babbitt metal, magnolia metal, Parsons and others, with varied claims for anti-friction and endurance qualities. The proportions for babbitt metal are: tin, 10 parts; copper, 1 part, and anti-mony, 1 part, by weight.

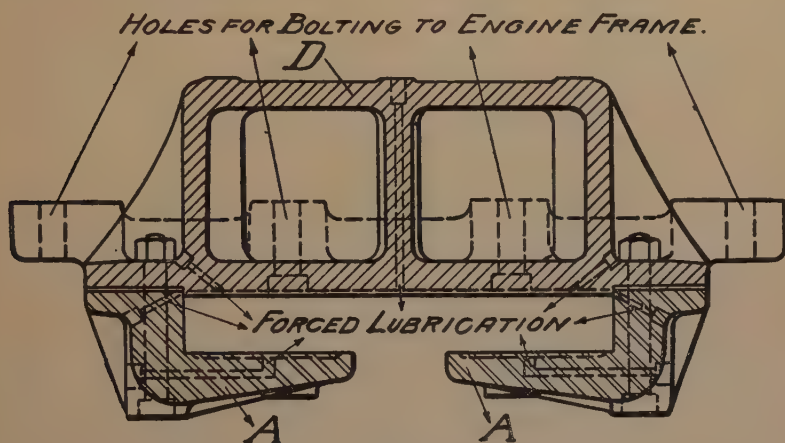


FIG. 70.

Another type of crosshead in common use is shown in Fig. 71 and is forged on the lower end of the piston rod. It is fitted with cap *A*, bearing brasses *BB*, forming the bearing for the pin situated at the upper end of the connecting rod, which in this case is of the type shown in Fig. 2, Plate 13. The slipper is secured as in the previous type, and the lugs for the backing guide in the same manner as before.

Connecting Rod and Crank.—The connecting rod is the link connecting the crosshead with the crank pin, by means of which

the reciprocating motion of the former is transformed into the circular motion of the latter.

Turning Force.—The force transmitted by the connecting rod to the crank pin may be resolved into two forces at right angles to each other, one component acting along the line of the crank and the other at right angles to it (tangentially to the crank pin circle). The component along the crank simply exerts a pressure on the shaft bearings, while the other is the component causing the crank to revolve, and is known as the *turning force*. These components vary constantly in magnitude with each other during the stroke,

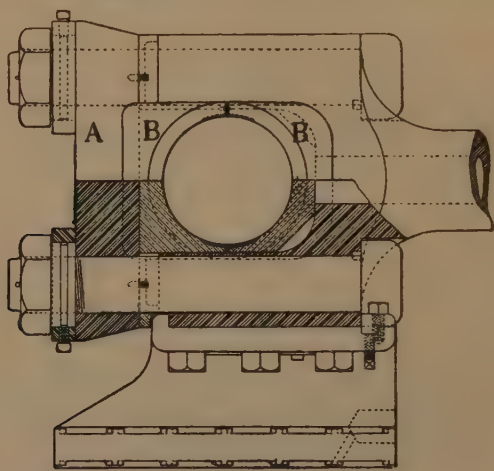


FIG. 71.

for instance, at the dead points, the tangential component disappears and there is no tendency to turn the crank, while the pressure along the crank is at a maximum. When the crank reaches such a position that the direction of the connecting rod makes a tangent to the crank pin circle (Fig. 72), then the radial component becomes zero while the tangential component or turning force is at a maximum. This position is obtained twice in a revolution and depends upon the angle θ made by the connecting rod and line of piston rod motion, and, consequently, upon the length of the connecting rod compared with the crank. If the former is infinite in length, it would become tangent to the crank circle

when the crank stood at right angles to the piston rod line of motion, while the shorter the connecting rod the nearer will this point of tangency of position of maximum turning force approach the dead center. For a ratio of $\frac{\text{connecting rod}}{\text{crank}} = r$, the maximum value of the turning force with a force P acting on the piston would be:

maximum turning force $= P \sec \theta = P \frac{AC}{AB} = P \frac{\sqrt{r^2 + 1}}{r}$. This irregularity of the turning force causes increased stresses in the shafting and tends to produce an uneven rotation, and becomes greater if the steam is cut off early in the stroke. Though the mean tangential pressure on the crank pin is less than the mean pressure driving the piston, there is no loss in the employment of

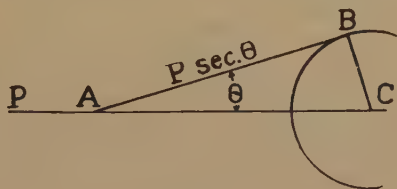


FIG. 72.

the crank, for the work transmitted in each case is equal to the pressure multiplied by the distance through which it moves, the work of the greater mean piston pressure moving along the diameter is equal to that of the lesser tangential passing over the circumference. The *dead points* or *dead centers* of the crank are the two positions where the crank lies in the direction of the line of piston stroke.

The connecting rod is also subject to stresses due to the inertia of the reciprocating parts, which in high-speed engines are of considerable importance and are discussed in Chapter XI. The effect of the *whipping motion* of the rod is also one entering largely in the design, and is due to the additional bending stresses set up by the transverse inertia on the connecting rod itself. These stresses increase rapidly with the speed of the engine and are a maximum in that part of the rod near the crank end, where the cross motion

is greatest. This is provided for by distributing the weight so that the resistance to the bending stresses may be at a maximum at the part where the stresses are the greatest. For instance, the rod is usually tapered from the crosshead end towards the crank end and is either given a rectangular section, or, as is the common practice in naval work, the rods are tapered and then planed away on the sides, the greater amount of metal being left in the plane of motion.

Connecting rods for the navy are made of high-grade steel forgings and hollow, a hole being bored from one end to the other. Figs. 1 and 2, Plate 13, illustrate the types of connecting rods in general use. The former is the type employed in the navy almost exclusively. This rod has a T end at the bottom, with a forked end at the top embracing the crosshead pins shown in Fig. 68, and fitted with adjustable bearings and connections. Fig. 1, Plate 14, shows the most recent design, fitted with forced lubrication.

The length of the connecting rod is measured from the center of the crank pin to the center line of the crosshead pins, and in the naval service is about twice the length of the engine stroke, or, as it is usually expressed,

$$\frac{\text{Length of crank}}{\text{Length of connecting rod}} = \frac{1}{4}.$$

For the bearing at the crank end and for each bearing at the crosshead there are two brasses with tin liners (to effect adjustments) and steel caps. The parts are connected by means of through bolts, and each bolt is fitted with a collar nut, with a set screw for locking the nut, and one for holding the bolt in place while backing off the nut. Formerly distance pieces (Fig. 3, Plate 13) were fitted between the brasses and held in place by small dowel pins. They can be withdrawn when the rod needs adjustment, without taking down the whole end, by slacking back the nut sufficiently to free the distance pieces from the dowel pins. Recent designs, Plate 14, omit distance pieces.

Two styles of cap bolts in general use are shown in Fig. 73, and designed to make the bolts of uniform strength along their length. In a crank revolution the bolts are first stretched and then returned to their normal condition, and, unless otherwise provided for, this stress will concentrate on the weakest part of the bolt, viz., the section at the bottom of the thread under the nut, and ultimately the bolts will tend to break at this point. To counteract

this tendency the solid bolt is turned down in two places so that the cross area remaining is about equal to the area of the threaded part at the bottom of the threads. The reduced bolts are of fairly uniform strength along their length, and are stronger to resist the shock coming suddenly on them at the bottom of the stroke when the tension comes at the beginning of the up stroke.

The second form of bolt accomplishes the purpose of making the bolt of uniform strength throughout its length by means of the hole through the axis.

The second type of connecting rod (Fig. 2, Plate 13) is forked similarly to the type described above, but instead of having brasses to fit the outside crosshead pins, has a *gudgeon* or wrist pin between the jaws to work in brasses in the end of the piston rod.

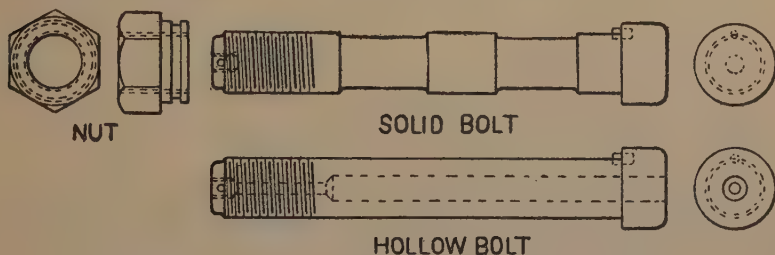


FIG. 73.

This type of connecting rod end is formed to suit the type of crosshead illustrated in Fig. 71. This shows also another method of forming the crank-pin end; this style is largely employed in English naval vessels and in the mercantile marine. The head of the rod is forged solid, the rod and cap being turned in the same operation, and then the hole for the brass slotted out roughly. The head is slotted through to cut off the cap, and the latter then bolted to the end of the rod with distance pieces interposed, and the hole bored out for the brasses.

Crank Shafts.—For naval purposes crank shafts are forged solid, then bored; being difficult to forge large shafts with several cranks, they are usually made in sections containing one crank each, as in the three-crank triple-expansion engines in use in our navy, or in two-crank sections for four-cylinder engines. Each section has couplings forged on the ends and the sections are secured together by coupling bolts and nuts. The sections are made symmetrical so that they are interchangeable, and a spare

section is sufficient for the replacement of any section disabled through accident. The crank shaft is made of high-grade steel forgings and the shaft and crank pins are each hollow from end to end, the diameter of the axial hole being about 60 per cent of the outside diameter.

The advantages of hollow shafts, pins, etc., may be expressed as follows:

1. A greater strength may be obtained with less material than would be necessary with a solid shaft, the saving in weight amounting to about 15 or 20 per cent.

2. The central portion removed is the least effective to resist twisting, and is the part most liable to contain defects or flaws, while the increased outside diameter necessary is very slight.

3. The axial hole offers opportunity for more extended inspection and increases the possibility of detecting flaws in the material.

Strength of Hollow Shafting.—The hollow shaft is therefore much stiffer against bending by its own weight, but is more costly of manufacture than the shaft solid throughout.

Let d = diameter of a solid shaft.

d_1 = diameter of a hollow shaft whose internal diameter is d_0 .

As shown in the text-books on *Applied Mechanics*, the moment of resistance of the solid shaft = $\frac{\pi f d^3}{16}$ and the moment of resist-

ance of the hollow shaft = $\frac{\pi f (d_1^4 - d_0^4)}{16 d_1}$. For equal strength,

$$\frac{f d^3}{5.1} = \frac{f (d_1^4 - d_0^4)}{5.1 \times d_1}; \text{ or } d^3 = \frac{d_1^4 - d_0^4}{d_1}.$$

Making the internal diameter half that of the external diameter and substituting in the above

$$d^3 = d_1^3 \times \frac{15}{16}; \text{ or } d_1 = d \sqrt[3]{\frac{16}{15}} = 1.022d.$$

Or the diameter of a hollow shaft to equal a solid one in strength is required to be only 2 per cent larger with a hole equal to half the diameter.

Fig. 1, Plate 15, represents the three-throw cranks of a triple-expansion engine used in the navy. *AA* are the *coupling discs*,

BB are the *crank shaft journals*. A *journal* is that part of a shaft or pin or rotating piece fitting into and working in the bearing. A *bearing* is that part of the casting or framing supporting the shaft and contains *bearing brasses* or *boxes*. When the brasses are made in two parts the line of division should be at right angles to the direction of the maximum thrust. The crank webs are at *CC* and in naval practice are chamfered as shown; *DD* are the crank pins. The axial holes in the crank pins are utilized for extra lubrication, one end is closed by a brass plate, the other end being fitted with a centrifugal oiling device.

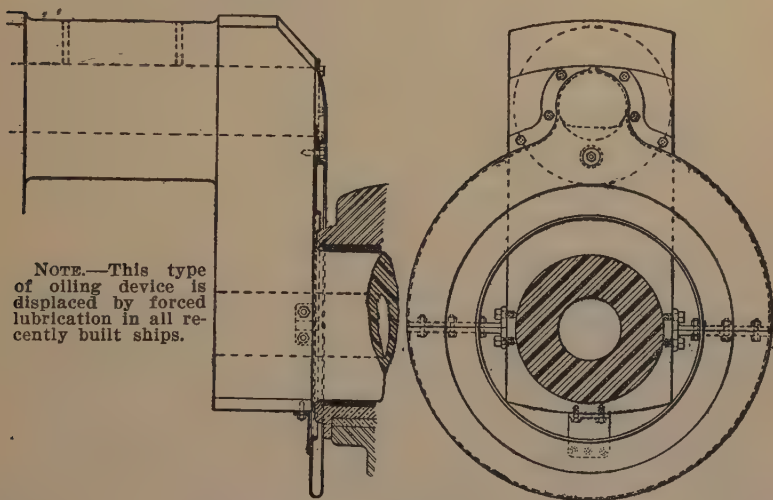


FIG. 74.

The *Centrifugal Oiler* consists of a copper or brass annular disc secured to the outside of the crank web and connected to the hollow pin which has a number of small holes leading from it to the bearing surface of the pin. As the shaft revolves the oil is thrown from the center by the centrifugal action, and makes its way through the holes mentioned to the surface of the pin. Oil is supplied by an ordinary drip delivering inside the annular disc close to the bearing.

The crank-pin journals require careful attention, as they are subject to varying stresses. The crank-pin brasses and main bearing brasses are all lined with anti-friction metal, and every care is taken to insure proper lubrication. Distance pieces and tin liners

are fitted between the brasses for adjustment as in the cases previously mentioned, and in all places where there are adjustable brasses. As the brasses wear and the clearance between the brasses and the journal becomes too great, the bolts are slacked off, two or three pieces of soft lead wire, about No. 16 B. W. G., are inserted between the brasses and journal, the brasses tightened up and the wire flattened, showing the amount of play and to what extent the thickness of the liner must be reduced.

The *built-up* crank shaft is represented in Fig. 2, Plate 15, and differs from the *solid* crank shaft from the fact that shaft, webs and pins are all separately forged and then machined, fitted and secured by shrinking and keying the various parts together. This method is extensively used in mercantile marine practice.

Fig. 1, Plates 16 and 17, shows a recent design with radial holes *A* in shaft and crank pins leading to the bearing surfaces, and holes *B* in the crank webs connecting the shaft axial hole with the crank pin hole. Brass plates *C* close the axial holes in pins and in the ends of shaft. As described under Forced Lubrication, the shaft hole is utilized for supplying oil under pressure.

For simplicity and uniformity, the crank shaft is made the same size from end to end, although the after section is subjected to a more severe stress than the forward cranks, as it must transmit the power of three cylinders in the case of a triple-expansion engine, while the forward section simply transmits the power of its own cylinder.

Bed Plates, Engine Framing, Columns.—The *engine bearers* or *seatings* are a part of the structure of the ship used to support the engine, and consist of a series of transverse and longitudinal steel plates built up from the inner bottom of the ship, stiffened and connected by angle irons, and built into the structure of the ship, as shown in Fig. 2, Plate 18. The *bed plate* forms the bottom part of the framing and is used to support the columns and carry the weight of the engine. As fitted in the larger naval vessels, it usually consists of sets of cast steel longitudinal and transverse girders of I section well stiffened by ribs and made in sections with flanges for bolting together, as shown in Fig. 1, Plate 18, and forming one rigid structure, the transverse girders carrying the main bearings of the crank shaft.

The bed plates *FF* are made of cast steel and in this case of three sections, bolted together and shaped for strength to fit the

engine seatings; the castings are strengthened with ribs, shaped to receive the crank-shaft brasses *AA*, and faced to fit the feet of the engine columns *BB*.

The *columns* support the cylinders and connect them to the bed plate, suitable feet being cast on the bottom of the cylinder and facings on the bed plate, and the whole rigidly connected together, with facings on the back columns to receive the crosshead guides. In the navy there are two forms of framing used:

1. The inverted Y cast steel back column, fitted to carry the crosshead guide and steel cylindrical columns in front of cylinder.

2. Cylindrically forged steel columns back and front, with athwartship tie rods and with horizontal and diagonal tie rods secured to the columns for rigidity and strength of structure, to-

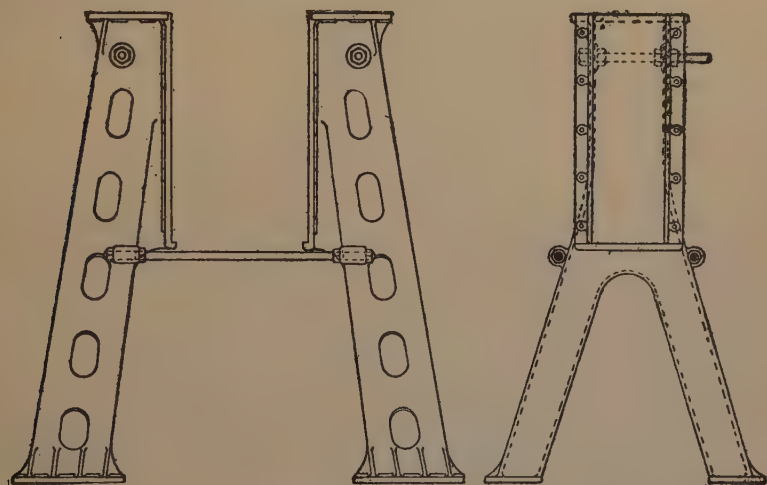


FIG. 75.

gether with supporting bars on the back columns for holding the crosshead guides. This form is generally used now.

Fig. 1, Plate 19, shows the first method, and Fig. 2, Plate 19, shows three views of the second method.

All columns are stiffened by athwartship horizontal steel tie rods *GG* and by the fore-and-aft horizontal rods *K*, while *racking* athwartship is prevented by the diagonal steel stays *G'G'*, and in the fore-and-aft direction by other diagonal stays, *M*. Provision is made for adjusting the stays by the fitting pieces shown.

A section of the crosshead guides is shown at *C*, and a full view at *E*. Each guide is bolted at the upper end to the cylinder, and at the lower end to a cast steel bracket, which is in turn bolted to the engine columns.

Crank-shaft Bearings.—The large bearings used for the shafting of marine engines should be made so that they may be readily removed for adjustment as wear occurs and thus prevent distortion of the shaft. In order to obtain this end, the bottom crank-shaft brasses are usually made concentric with the shaft so that they may be revolved and taken out without removing the crank shaft, as represented in Fig. 2, Plate 14, showing one of the crank-shaft bearings of the Oklahoma class. The bed plate is recessed to receive the bottom brass, which is round at the bottom and can be removed by supporting the shaft and revolving the brass in its cylindrical bed until it comes to the top, and so be taken off for adjustment or repair. The bearing has a cap of cast steel; the part coming next to the shaft, as well as the bottom brass, is lined with white metal as previously described, and fitted by scraping. Each cap is secured by bolts, provision being made to prevent the nuts from backing off and the bolts from turning.

Each bearing consists of a shoe *A*, fitting in the bed plate, a bottom brass *B*, a cast steel cap *C*, and adjusting liners *D*. To prevent the round brass from nipping the shaft when heated, it is shouldered against the steel cap as shown. The cap and bottom brass are cored to permit circulating water through them. The cap is secured by four collar bolts, the nuts locked to prevent their backing off, and the bolts prevented from turning by set bolts. Note the holes *E*, and grooves in white metal *F* for forced lubrication. Many older bearings still in use have distance pieces similar to Fig. 3, Plate 13, between the cap and bottom brass, and an oval hand-hole in the cap to permit feeling the journal, but lack the holes *E* and grooves *EF* for forced lubrication.

Line, Thrust and Propeller Shafts.—Fig. 2, Plate 16, gives a line sketch of the shafting in a twin-screw ship. From the crank shaft the motion is communicated to the propeller by means of sections of shafting carried in bearings and secured to each other by flange couplings. The after section carrying the propeller is known as the *propeller* or *tail shaft*, the section fitted to work in the thrust bearing to transmit the thrust of the propeller to the ship is called

the *thrust shaft*, the section in the stern tube the *stern tube shaft*, and the intermediate sections are known as the *line shafting*. The shafting is of high-grade steel forging, and hollow from end to end for the reasons stated before.

The line shafting is free from the bending stresses brought on the crank shaft and has only to transmit the torsional stresses due to the revolving screw propeller, so that the part between the crank shaft and the stern tube can be reduced in diameter. On this account it is customary to provide for bearings on this part of the shaft, called *spring bearings* (Fig. 76), consisting of a pedestal or pillow block fitted with a bottom brass only and fitted

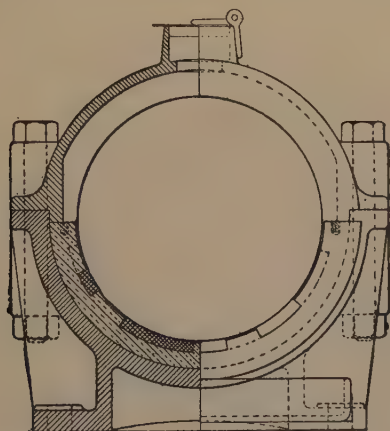


FIG. 76.

as usual with white metal lining. A cap or cover is sometimes fitted, not in contact with the shaft, to protect the journal from dirt or to hold an oil box. From the stern tube and beyond, the shaft is increased in diameter, usually to that of the crank shaft, to avoid the danger of breakage in the stern tube and to ensure extra stiffness outside.

The shaft immediately abaft the crank shaft is constructed with a number of collars on the journal to bear against corresponding surfaces in the *thrust bearing* and thus transmit the thrust of the propeller to the ship, the pressure being exerted on one side of the collars in ahead motion and on the reverse side in going astern. The

bearing is secured firmly to a bearer or seating rigidly connected to the structure of the ship and extending over several frames.

The usual form of thrust bearings for all modern naval vessels, and known as the *horseshoe collar thrust*, from the shape of the bearing rings, is shown in Fig. 1, Plate 22. As applied to the Louisiana, it consists of eleven collars turned on the shaft with the same number of thrust collars in the bearing. The bearing consists of a *pedestal*, *A*, bolted to the *sole plate*, *B*, with holes in the former, so that as wear takes place on the collars the whole block can be adjusted bodily by means of wedges, *C*, bearing against

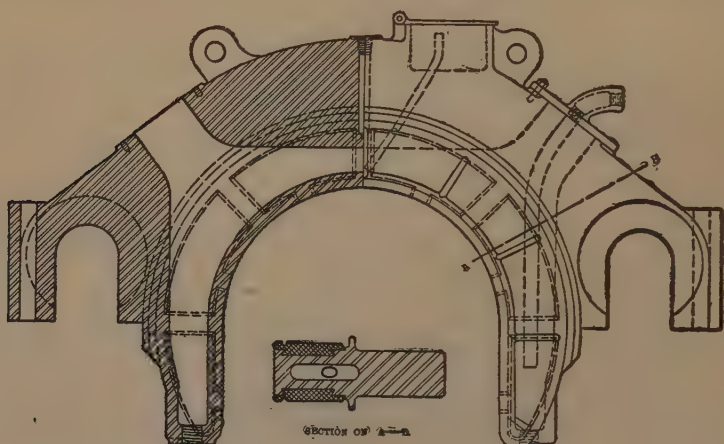


FIG. 77.

lips on the *sole plate*. Means of adjustment are necessary, otherwise the shaft will gradually wear forward, throwing the cranks out of line and bringing undue stresses on the engine. The *pedestal* is of cast iron, the ends and side walls forming a trough partially filled with oil in which the shaft collars revolve.

There is a *steady bearing* *D*, at each end, cast in one with the pedestal and relieves the thrust bearing of the weight of the shaft, so that its office is only that of taking the thrust of the propeller. Each steady bearing is fitted with a bottom brass and cap, both lined with white metal, and at the forward end of forward, and the after end of after steady bearing there is a *divided stuffing box*

and *gland* to prevent the escape of oil. The oil trough in the pedestal contains a circulating pipe or coil for cooling water to reduce the temperature of the oil and bearing.

The *thrust bearing collars*, *E-E*, shaped as shown in detailed sketch, Fig. 77, are inserted between the shaft collars and all secured to the two side rods, *F-F*, threaded throughout their length and carried by the pedestal, one on each side. The holes at the extremities of the horseshoes for these bolts are slotted through, so that each horseshoe may be taken out independently of the others for repairs, and each is fixed in position on the side rods by a shallow nut on each side of the collar, thus securing individual adjustment of each ring. The horseshoes are still further secured or steadied by bolts *G*, to keep them from rising. They are made of cast steel, cored for lightness and faced on each side with white metal and channeled for distributing the oil, and each one provided with an oil box on top with tubes leading to both collars.

The diameter and number of thrust collars are determined by first calculating the thrust of the propeller, to be explained later, and then determining the total thrust bearing on a basis that the pressure per square inch from the thrust should not exceed about 70 pounds per square inch on the bearing surface.

Another type of thrust bearing much used, is known as the *collar thrust bearing*, shown in Fig. 2, Plate 22, as applied on the Raleigh. This consists of a *pedestal A*, *sole plate B*, *lower part* of the bearing and the *cap D*. The lower part of the bearing is of cast iron with internal collars to alternate with the shaft collars which press upon the after or forward faces of the internal collars, as the screw is going ahead or backing, as in the horseshoe type. The cap is filleted to fit the upper flange of the pedestal and secured to it by bolts, *E*, body bound in the pedestal, and contains internal alternate rings to correspond with those in the bottom part. The thrust bearing rings are lined with white metal, channeled or grooved, and take the thrust of the shaft, as before, from the shaft collars. The ends and side walls of the pedestal form an oil trough, *F*, from which there leads an oil hole to each collar and recess. Inside of the oil trough, forward and abaft the thrust collars, there is a steady bearing, *G*, fitted with a divided stuffing

box and gland as in the preceding type, and at the bottom a channel is formed connecting all the bearing recesses with a drain cock at each end for emptying the oil chamber.

The bearing can be adjusted bodily in a vertical direction by means of wedges and set screws under the steady bearings shown at *H*, and in a fore-and-aft direction by two set screws, *K*, at each end, bearing on the thrust block.

A longitudinal oil box is cast on the cap and fitted with a hinged cover, while projecting lugs on the cap and pedestal fit snugly together and take the end thrust of the cap.

Stern Tube Bearing and Shaft.—As the shaft leaves the ship and enters the water a bearing of special construction becomes necessary owing to its being under water, its inaccessibility and

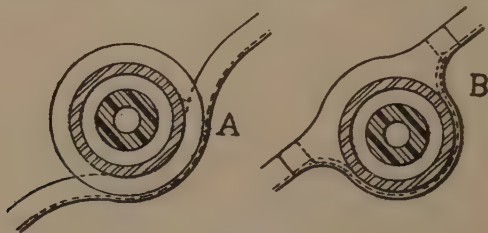


FIG. 78.

the necessity for properly supporting the length of the shaft. The details of a *stern tube bearing* are shown in Plates 20 and 21, and consist of the shipbuilders tube Fig. 1, of steel worked into the frames during construction and secured to the ship by a flange on the inboard end. At the ends steel *bushings* or *rings* are secured and bored out to receive the *stern tube bushing* or *lignum vitae lined bearing*. In twin-screw ships the *frames* and *plating* are bossed out where the shaft leaves the ship so as to carry the stern tube well aft, as shown in the sections *A* and *B*, Fig. 78. The stern bearing is usually in two parts, one at the inboard end of the stern tube and the other at the outboard, and consists of a *composition sleeve*, grooved to receive dovetailed strips of *lignum vitae* inserted longitudinally in the interior of the bushing, and set so as to present the end of the grain to the shaft. The bearing is

lubricated by sea water circulating through the channels, filling the space between the stern tube and *shaft casing* and flowing through to the inboard end. The flow of water into the ship at the inboard end is prevented by means of the *stern tube stuffing box*, Fig. 3, Plates 20 and 21, bolted to the flange on the forward end of stern tube bushing, and is made in halves divided longitudinally. Underway, the glands can be slackened back to permit circulation of water through the stern tube, and in addition there is a cock through which water may be drawn to judge by its temperature the condition of the bearing.

The bushings are made in halves, the joints being in a horizontal plane when they are in place so that they may be easily withdrawn for needed repairs, and the lignum vitae is well water-soaked and bored out in place to perfect alignment, and to a loose fit on the shaft casing.

Stern Shaft Casing.—That portion of the stern shaft revolving within the stern tube is covered with a composition water-tight casing to protect it from the sea water and to furnish a good bearing material on the lignum vitae. This casing may be fitted in various ways; usually after turning and boring, it is forced onto the shaft by hydraulic pressure and further secured by studs in the casing and shaft. Where possible, the casing should be in one length, as the greatest care is necessary to prevent any leakage of water between the casing and shaft, or the shaft would be seriously corroded by galvanic action at the joints. In our navy for large shafts, the casing is usually made in sections, shrunk and pinned on, about one inch in thickness at the bearings and a half inch elsewhere. The joints are filleted with a lap of about one inch in length as shown, and the joint itself sealed on the outside with soft solder. Details of the joint are shown in Fig. 6, Plates 20 and 21.

Strut Bearings.—The after part of the propeller or tail shaft works in bearings near the propellers carried by struts rigidly secured to the ship's structure. The struts are bored in place, concentric with the center line of the shaft to seat the bearings which are made of composition, in halves, and fitted with lignum vitae strips similarly to the stern tube bearings. The bushings are

secured to the forward end of the strut by flanges and the lignum vitæ held in place by flat rings.

The shaft is also covered with a casing at the strut bearing secured similarly to that on the stern bearings and extends from the forward end of the fair water to one inch within the propeller hub, and the joint between the casing and propeller hub must be absolutely tight.

Zinc protective rings are fitted on the stern bearings and on the strut bearings to arrest the action on the hull and shafting, from the proximity of the two dissimilar metals.

For sheathed ships, when the bottom is coppered, it is necessary to cover the entire shafting under water with a composition casing similar to that employed for shafting in the stern tube, owing to the excessive galvanic action set up by the copper and exposed steel surfaces. To prevent action on the steel couplings under water, composition boxes or casings, in halves, and made water-tight similar to that in Fig. 3, Plate 16, are employed.

Fair Water Sleeves made of plate steel in halves are secured to the sleeves of the stern tube bearing and strut bearing to reduce the resistance to passing through the water.

Shaft Couplings.—The various lengths of shafting are secured to each other by means of flanged couplings, shown at A, Fig. 1, Plate 15. The flanges are forged solid with the shaft and connected by large snugly fitted bolts and secured by nuts. With interchangeable crank shaft sections, the number of bolts in the couplings must be a multiple of the number of cranks so that the bolt holes will correspond if a section is shifted, or a new one inserted. For instance, with three cranks at equal angles the number of coupling bolts would be 6 or 9 and for cranks at right angles 8 or 12 would be required. One exception to the above method of coupling is the method used in connecting the section of the shaft passing through the skin of the ship when there is a coupling between it and the propeller. In this case the inboard end must be made with a removable flange so that the shaft may be inserted or withdrawn from the outside and fitted from the inside. There are many methods of making this connection, the one in common use in our navy is shown in Fig. 4, Plates 20 and

21. The coupling flange at the after end of the section of shafting forward of the stern tube shaft is solid as shown at *A*. The forward end of the stern tube shaft *B* has a groove *C*, extending all around it. In this groove fit the pressure ring segments *D* and *D'*. Keys *E* assist in securing the coupling sleeve *F* to the shaft *B*, and take the torsional stress.

To assemble the coupling sleeve, *F* is placed on *B* and the keys driven home; the two flanges *F* and *A* are brought together, the pressure ring segments *D* and *D'* are inserted radially and the whole is firmly bound together by the coupling bolts *G*.

The upper right hand view is looking toward the shaft *B*, sleeve and segments assembled, with shaft *A* removed.

A form of outboard coupling frequently used is shown in Fig. 5, Plate 20. The ends of the two shaft sections, *A* and *B*, slightly tapered, are drawn into a sleeve *C* by means of heavy through keys *D*. At *E* is shown on a large scale in transverse section one of the longitudinal keys (*F* in the upper view) with set screw locking it. This coupling is designed to offer little resistance to the water and also to uncouple readily after long immersion in salt water.

The shaft casing commences just abaft the forward coupling and extends to one inch inside the propeller hub in vessels fitted with only one section of outboard shaft; this joint in the hub must be water tight. The casing is shrunk and pinned on. The joints between adjacent lengths of casing are scarfed, and, on the inside a $\frac{1}{4}$ " space is left which is pumped full of a mixture of red and white lead and boiled linseed oil to prevent galvanic action. At the ends where the joints between the casing and shaft are exposed, steel rings, made in halves, are secured together and the joint made water tight.

Forced Lubrication.—Oil supply under pressure to engine journals has been used on dynamos and small engines of high rotary speed for many years, and more recently has been fitted to turbine machinery. In the last few years it has come into general use for the principal bearings of reciprocating main engines, and even for much of the auxiliary machinery.

Considerable increase of efficiency, perhaps as much as 4 per cent at low power and somewhat less at full power, results, owing to marked decrease in the friction load.

Moreover, the wear on bearings is so much lessened as to effect great saving in expense and labor required for adjustments and repairs. The "burning out" of bearings, practically unknown with forced lubrication, might prove a fatal accident to a battleship. Again, very great saving in the use of oil results, which is a considerable item with ordinary lubricating appliances using from thirty to several hundred gallons of oil daily, depending upon I. H. P. developed and largely upon the care used.

Each main engine has a separate system.

To prevent loss of oil, crank pits are made oil tight and complete oil tight casings are fitted around the engine, open at the top, about five feet high.

Two tanks of several hundred gallons capacity each are located conveniently above each engine. One of these is the main oil supply tank, and contains a copper cooling coil through which sea water is circulated; the other is a settling tank and contains a copper steam heating coil. Oil after use is pumped into the settling tank where sediment and water are removed. This is accomplished readily by heating the oil to about 160° , and when thus purified it is pumped to the main supply tank.

Usually three pumps are installed; one to supply the bearings with oil, one to return oil from the oil well in crank pit to the supply tank or to the settling tank, and the third fitted as a spare to perform the work of either one of the others. Oil is supplied under from 10 to 25 pounds pressure, runs off the bearings freely and drops into the crank pit.

Fine mesh brass wire gauze strainers are fitted in discharge lines to the main bearings and to the tanks, and in the suction lines from the crank pit. The strainers are readily accessible for cleaning.

Thermometers are located at the main journals to show the temperature of oil in the bearings, and pressure gages are installed to show the oil pressure.

CHAPTER IX.

THE CONDENSER AND ITS FITTINGS.

Formation of Scale.—Sea water contains about $\frac{1}{32}$ part of its weight of solid matter, the average composition being:

" "	Sodium Chloride	76 per cent.
	Magnesia Chlorides	10 " "
	Magnesia Sulphate	6 " "
	Calcium Sulphate	5 " "
	Carbonates and Organic matter	3 " "

These salts become insoluble at definite temperatures, and are also precipitated by increased saturation due to evaporation, and when deposited on the surface of a boiler, form a non-conducting scale which not only causes a great waste of heat but rapidly burns out the heating surfaces, with the attendant collapse of furnaces or destruction of tubes. Of the constituents mentioned above, sulphate of lime is the most objectionable. The scale formed from sea water contains about 80 per cent of this salt, forms a hard deposit and one difficult to detach. Sodium chloride or common salt remains soluble at the temperatures used, and is not deposited from increased density until a saturation of $\frac{8}{32}$ is reached. Sulphate of magnesia contributes to the deposit but in a less objectionable form.

At a temperature of 280° to 295° F., corresponding to a pressure of 35 to 45 pounds of steam per gage, calcium sulphate becomes insoluble, and at 350° or 105 pounds per gage, all sulphates are deposited, so that at this temperature, all feed water containing these salts forms a precipitate immediately on entering the boiler.

The office of the *surface condenser* is to obtain the benefit of a partial vacuum on the engines and also to keep the water of condensation of the exhaust steam separate from the condensing or sea injection water, so that it can be used again in the boiler. By this means, any deposition on the boiler surfaces is prevented.

The earlier steam engines using low pressure steam were formerly fitted with *jet condensers* in which the steam exhausts into a chamber and is there condensed by a jet or spray of sea water. This mixture was but slightly fresher than sea water and was fed to the boilers, necessitating frequent blowing down, with the attendant loss of heat, in order to prevent increased density. This process was possible with the low pressures in use, but as the temperatures and pressures were increased it became impossible not only to prevent the deposit, but it was greatly increased, owing to the additional salt water supplied to make up the loss in blowing down. For this reason the jet condenser was abandoned for marine purposes and the surface condenser is used for all sea-going vessels.

The condenser sometimes forms part of the engine framing. In this case the shape is generally rectangular and the shell and water chambers are made of cast iron, while the other details are in general similar to that of the condenser separate from the main engine. Plate 23 shows such a condenser as applied on the Tonopah, the upper part being cast to the back column as shown, and the whole bolted to the bed plate and engine foundations.

This is an unusual type, rarely met with in naval ships.

The following openings, connections and attachments are fitted to all surface condensers:

A connection from the auxiliary steam pipe to the bottom man-hole for cleaning the tubes by boiling.

Drain cocks and pipes leading to the bilge for draining the condenser.

Air cocks at the highest parts of the water chests.

Spring safety valve loaded to 25 pounds pressure above the atmosphere.

A hand hole about six inches by three inches in the bottom of each tube sheet.

Area of Condensing Surface.—In the naval service, the allowance of cooling surface varies from 1 to 1.25 square feet per I. H. P. for battleships and cruisers, and in torpedo boats and destroyers, where reduction of weight is of great importance, this has been reduced to .85 square feet per I. H. P.

Vacuum Gage.—The vacuum gage is used to record the extent of the vacuum in the condenser, which is determined by the height of a column of mercury it will support. With a complete or perfect vacuum this would be about 30 inches. The vacuum usually attained in a condenser is from 26 to 28 inches, and the absolute pressure of the vapor in the condenser corresponding to 26 inches of vacuum, would be, since 2 inches of mercury are equivalent to one pound pressure,

$$\frac{30 - 26}{2} = 2 \text{ lbs.}$$

In the engine room the vacuum is indicated by a Bourdon gage similar in construction to the ordinary pressure gages. The indications are in inches, possibly from the fact that the old original vacuum gages were constructed like a barometer, and to avoid confusion when the spring gages came into use, they were marked in this manner.

In engine tests the barometric height should also be taken in order to determine the maximum attainable vacuum under the working conditions. For this purpose the temperature of the condenser is noted and the pressure corresponding to that temperature taken from the Steam Tables (see Appendix). Deducting this pressure from the atmospheric pressure as obtained from the barometer, the difference would be the maximum attainable vacuum with that temperature. Thus, if the temperature of the condenser is 101° F., and the barometer stand at 30 inches, equivalent to a pressure of 14.7 pounds per square inch, the vapor pressure corresponding to 101° is .980 pounds per square inch, and, consequently, the greatest vacuum possible would be 14.7 - .980 = 13.72 pounds, per square inch below the atmosphere, equivalent to 27.5 inches on the vacuum gage.

Surface Condensers.—Formerly a few surface condensers were made in which steam circulated through the tubes, and the cooling water around them; this type is obsolete in marine practice, cooling water passing through the tubes in all modern designs.

The most usual type of marine condenser consists of a cylindrical or oval shell separate from the main engines, having a flat head called a *tube sheet* at each end, and over these tube sheets are *water chests* secured to the shell of the condenser to provide for circulat-

ing the cooling water through the tubes. Exhaust steam enters near the top of the shell and circulates around the tubes, giving up heat to the cooling water. The condensed steam, with much entrained air, is drawn off by the *air pumps*. The *circulating pump* forces the cooling water through the tubes, and overboard through the *out-board delivery valve*.

Until the advent of the turbine the simple surface condenser which could maintain a vacuum of about 26" answered the requirements, since to expand steam lower than about 23" (which corresponds to about 25" in the condenser) means great increase in volume with consequent increase in the size of the L. P. cylinders. To increase the L. P. cylinders to the extent necessary for expansion much below 23" would be impracticable, hence for the reciprocating engine over 26" of vacuum in the condenser is wasteful.

Since to realize the best economy in the use of turbines requires the highest possible vacuum, their adoption has brought about a variety of improvements in condensers and air pumps with a view to obtaining better vacuum while leaving the water of condensation at the highest possible temperature, and with economy in weight, space and power.

Increase of cooling surface, adoption of improved methods of keeping air from leaking into exhaust spaces or becoming mixed with boiler feed water, new methods of arranging baffle plates inside condensers or dividing condensers into compartments so to drain off water condensed in the upper part without its being needlessly cooled by contact with the lower tubes, increased speed of cooling water in general direction across that of the exhaust steam, and better methods of pumping air and condensed steam from the condenser are the general lines along which improvements have been made.

Changes have also been made in condenser tubing. Instead of tubes being packed at both ends they are usually expanded into tube sheet at one end and fitted with a ferrule and packing at the other; the ferrule has no shoulder to prevent crawling, the tube being held securely at the expanded end. Half of the tubes are expanded at one end (the end at which circulating water enters tubes, so to offer less obstruction to circulation), and the other half at the other end. In smaller condensers tubes are often all expanded at both

ends, a slight curvature in the tubes permitting expansion and contraction. Many of our destroyers have condensers of this type. Tube spacing may be closer at the bottom of the condenser, too, since exhaust steam when it first enters requires, of course, a more spacious pathway than after it is partly condensed.

Surface Condensers, Cylindrical Type.—A recent design for a battleship is shown in Plates 24 and 25. Referring to Fig. 1, showing one of the main condensers for the U. S. S. Texas, the shell *A* is made of steel boiler plate $7/16$ inch thick, with longitudinal double butt joints, each butt strap made of one continuous piece. The shell is strengthened by circumferential T-bar stiffeners, *B*, and there are angle irons *C*, riveted around each end of the shell, forming flanges to which the tube sheets and water chests are secured. Two composition nozzles *D* are riveted to the upper part of the shell, having faced flanges *X* for the attachment of the main exhaust pipes, one each from the forward and after L. P. cylinders. To the forward nozzle is secured a faced flange (not shown) for the auxiliary exhaust, bleeder, and safety valve connections, while at the bottom and sides are riveted cast-steel flanges *E* for the air pump suction, *F* for make-up feed, *G* for cylinder drains, and *H* for a manhole plate, the cover of the last being fitted with a 1-inch connection *I* from the auxiliary steam line to allow cleaning the tubes and interior of shell by boiling. The water chests *W* are cast of composition, the after chest being the one for the inlet and outlet of circulating water; the head of the latter chest is provided with a division plate *K*, which separates the space enclosed below it between the chest and tube sheet from that above, and this is fitted with valves which, when open, will allow the circulating water to pass through the division plate and overboard without going through the tubes; these valves are worked by a lever on the outside of the condenser. This water chest is also fitted with nozzles for circulating water inlet *L*, and outlet *M*. The water chest at the other end of the condenser is similar but has no division plate nor circulating water nozzles. Both water chests are provided with 10-inch manholes, fitted with composition covers *N*, to the inside of which are secured rolled zinc plate protectors *O* to minimize galvanic action between the composition and steel parts. The

chests and tube sheets are stayed by nine stay-bolts such as *P* and *P'*; these stay-bolts pass through the tube sheet and are secured into the steel stay tubes *Q* (method of staying is shown in Fig. 2). The tube sheets *R* are rolled of composition, each in one piece. They are $1\frac{1}{8}$ inch thick where tubes are fitted with ferrules, and $1\frac{3}{8}$ inch thick where tubes are expanded. The methods of fitting the tube ends are shown in Fig. 3. In the top and bottom of each tube sheet are hand holes and covers *S*, for the inspection of the tubes. There are 5540 seamless drawn composition tubes in this condenser, $\frac{5}{8}$ inch outside diameter, .065 inch in thickness, the tubes are 15 feet $2\frac{1}{2}$ inches long, and spaced 1 inch between centers; they are supported at two intermediate points by brass thimbles $\frac{3}{4}$ inch long, inserted in steel supporting plates *T*, $\frac{1}{4}$ inch thick as shown in Fig. 4. Each tube sheet and chest is secured to the flanges of the shell by through bolts *Y*, a number of which are provided with collars, Fig. 5, to serve for holding the tube sheet in place and retaining its joint with the flange on the shell when the outer joint is broken in removing the water chests to replace tubes.

The circulation of water and steam is as follows, viz.: The circulating water drawn from the sea through the main injection valve by the circulating pump, is driven into the lower half of the after water chest at *L* in Fig. 1; the diaphragm or division plate *K* causes the water to pass through the lower half of the tubes to the forward chest, and, filling that chest, the water returns through the upper half of the tubes to the upper half of the after chest, and is thence discharged overboard through the discharge pipe and the out-board delivery valve.

The exhaust steam enters the upper part of the condenser shell through the nozzles *D*, falls upon a perforated scattering plate *U*, and is distributed over the whole length of the condenser, and then to the body of the tubes; three sheet-steel baffle plates *V*, *V'*, *V''*, are fitted between the tubes, as shown in Fig. 1, by which the steam is directed equally over all of the cooling surface.

Baffle plates *V* and *V'* being inclined and having holes with raised edges, the water condensed above them is drained off toward the shell, and down outside the tubes, thus avoiding unnecessary cooling of the water, while the remaining steam and air is still

further condensed in its passage over the entire cooling surface, finally falling into the air pump suction where it is removed by the air pump.



FIG. 79.

Weir-Uniflux Condenser.—This type of condenser, shown in Fig. 79, depends for its efficiency on the contour and the tube spacing, the contraction of which toward the bottom causes a nearly uniform speed of the mixture of vapor, air, etc., over the

cooling surface from the exhaust inlet to its arrival at the air pump suction. It should be noted that by improper baffling or tube spacing in any condenser a good vacuum may be obtained throughout the *lower part* of the condenser, with a *poor* vacuum at the *exhaust inlet*—and that the latter place is where the good vacuum is desired; too close spacing of tubes would have the same effect as an exhaust inlet of insufficient size.

There are no internal baffle plates in the Weir-Uniflux Condenser. For the best results a dual air pump system—that is, a combination of two pumps, one of which removes dry air, the other wet air and water—is required, and this will be described later.

Contraflo Kinetic Condenser.—Figure 80 shows an outline sketch of this type of condenser developed in England, and used on some of the latest British ships. The exhaust inlet has a number of directing plates *A* so curved as to guide the entering steam and prevent the formation of eddy currents. Baffle plates *B* assist in forming a wedge pointing toward the air pump suction. The theory is that the wedge shape and the guide plates, forming several converging condensing compartments, tend to enable the cooled vapor with the entrained air to move directly toward the air pump suction, the cross sectional area of the stream lessening as the steam gradually condenses, thus causing the dry air pump to handle a comparatively dense medium, rich in air and of low temperature; it is also believed that the rush of exhaust steam guided by *A* assists in driving the vapor toward the air pump suction. The spacing of the tubes is also closer in the lower part than in the upper part of this condenser; and much of the condensed steam is drawn off by the sides and the baffles without being further cooled by contact with the lower tubes.

This condenser is similar to the uniflux. In both these condensers a division of air and water takes place, in the uniflux after leaving the condenser. This separation of water and air is aimed at in most all modern condensing plants where very high vacuum is required. It is clear that to pump air at a high temperature is not economical, since the colder it is the more dense, and the more readily removed by the air pump. On the other hand, the colder the water of condensation the greater the *loss in heat* passed overboard in the circulating water, whereas the higher its temperature

the greater the quantity of heat retained and returned to the boiler in the feed water. In the dual air pump system, or any equivalent method, the wet pump should handle water at near the steam temperature, while the dry pump aims to deal with air and vapor as dense and at as low a temperature as the temperature of the injection water will permit.

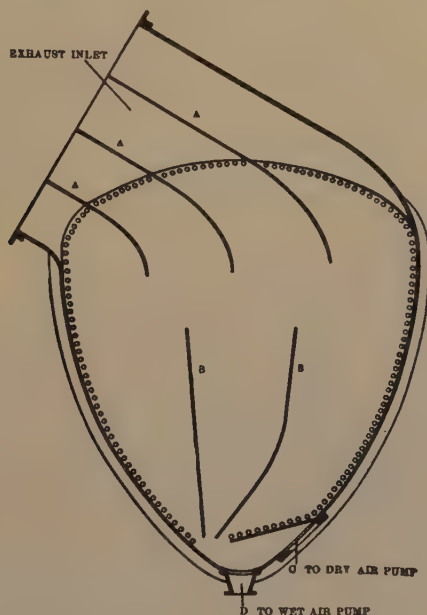


FIG. 80.

It has been found that the presence of air in a condenser in even minute quantities exercises an injurious effect in lowering the vacuum, far beyond that due simply to its direct increase of pressure—it exercises a marked effect in *lowering the heat transmission*. In other words, the presence of air seems to surround the tubes with a non-conducting medium, so that when present in considerable quantities the lowermost tubes become so air drowned that they will hardly condense any appreciable quantity of steam. This bad effect is minimized by keeping up active circulation of vapor over the cooling surfaces.

Air Pump.—The office of the air pump is to remove from the condenser the water of condensation together with the small quantity of air entering with the steam, or due to leakage, and which would otherwise destroy the vacuum.

The vertical air pump, illustrated in Fig. 81, is the one in general use for reciprocating engines in the navy and the mercantile marine. It consists of twin pump barrels in which work *pump buckets*, 83, fitted with metallic non-return valves, working in composition cylinder, 66, fitted at the bottom with the *foot* or *suction* valves, and at the top with the *head* or *delivery* valves. These valves all have a vertical lift of about $\frac{1}{2}$ inch. The bucket works with a very close fit in the barrel and is not packed except with water grooves shown in the cut. The buckets are driven by inverted vertical steam cylinders as shown. There is often only one steam cylinder instead of two. The air pump suction leads to the bottom of the condenser, except in recent designs where there are two air pump suctions, one leading to the bottom and one, to take dry air, leading to a point higher up. The discharge pipe leads to the *feed tanks*.

The operation of the pump is as follows: As the pump bucket rises a partial vacuum is formed between its lower face and the foot valves, causing the foot valves to open from the slightest increase of pressure in the condenser, and the air, vapor and water enter the pump barrel until the end of the up stroke. As the bucket descends the pressure closes the foot valves, and the water is forced up through the bucket valves to the space above. On the next up stroke this air and water above the bucket are discharged through the delivery valves, 77, to the hot well, and at the same time another supply enters the barrel from the condenser through the foot valves. The pump is consequently single-acting, the delivery occurring only on the up stroke.

Since the pressure in the condenser cannot be reduced below that necessary to lift the foot valves, the amount of vacuum that can be obtained in the condenser will depend upon the weight of the foot valves and the amount of vacuum created in the pump barrel itself. With the buckets working air tight, the vacuum attainable above the foot valves depends very largely upon the

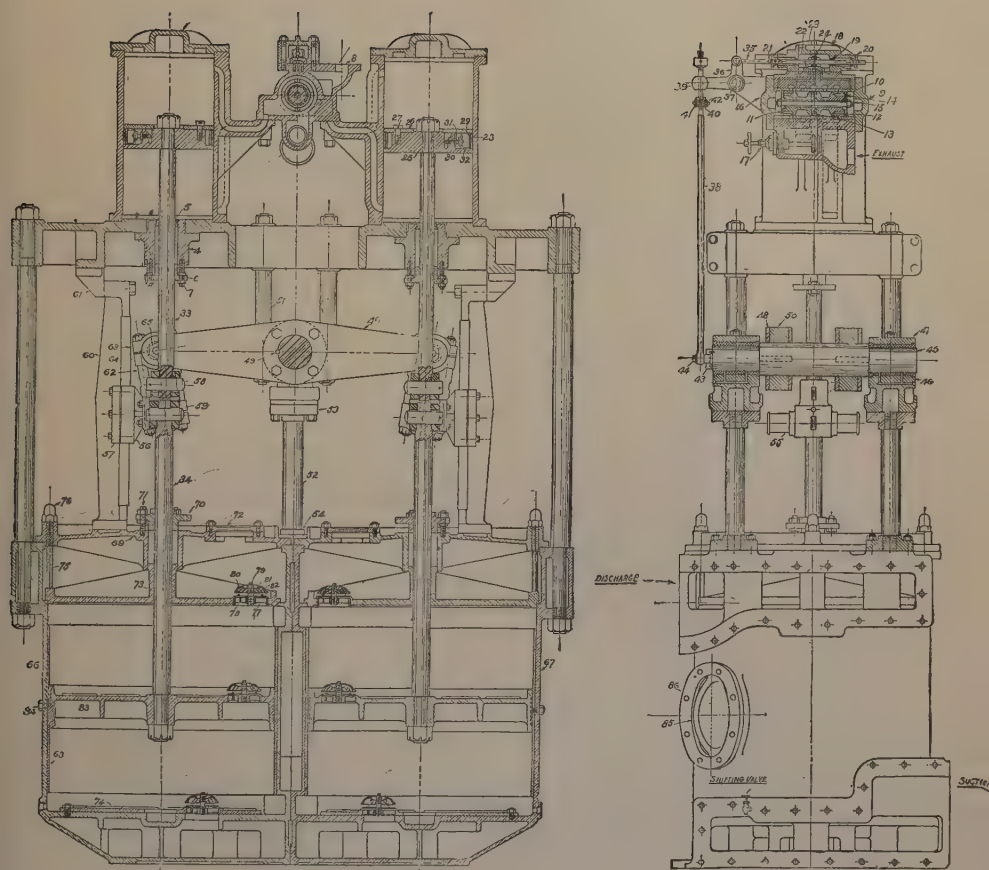


FIG. 81.

Blake & Knowles Vertical Single Acting Twin Beam Air Pumps.

clearance space between the bucket and the foot and delivery valves at the ends of the stroke. The effect of this clearance is to reduce the efficiency of the pump, or the amount of vacuum which the pump can create, so that in well-designed pumps, the clearance must be reduced to a minimum. All pockets where air can collect without being discharged at the end of each stroke should be avoided and the clearance found necessary should be kept filled with water for the best results. The efficiency is increased, especially with the double-acting horizontal air pump, by keeping the head valves covered with water. This is secured in the vertical pump by means of the ledge or dam shown in the figure.

As there is not much air present when the water reaches the head valves on the up stroke, and the bucket reaches the water on the down stroke, there is generally quite a shock or blow at these intervals. To overcome this, small cocks are fitted to the ends of the barrel and connected to a pipe with a check valve opening inward, permitting a small quantity of air to enter, thus acting as a cushion and so reducing the shock.

The speed of the pump has a decided effect upon its efficiency, an excessive speed resulting in churning up the water and air and giving a poor vacuum. In the merchant marine, the speed is from 200 to 350 feet per minute, while in the navy, piston speeds as high as 400 and 500 feet per minute have given very fair results.

The air pump is often driven by an attachment to the main propelling engines; but usually by an independent engine. When attached to the main engine the pump is operated by an *air pump lever*, usually taking its motion from the low pressure crosshead as in Plate 23, showing the air pump connections for the Ozark-Tallahassee class. The stroke of the pump is reduced to about one-third that of the main engine by the beam connection. In this pump instead of the head valve plate being secured to the pump casing, as in Fig. 81, it is held down by a stiff helical spring strong enough under normal conditions to resist the pressure of the water as it is forced up, but, if the pump becomes flooded with water, instead of the valves being carried away, the entire head plate will be lifted bodily and give a free discharge to the contents of the barrel.

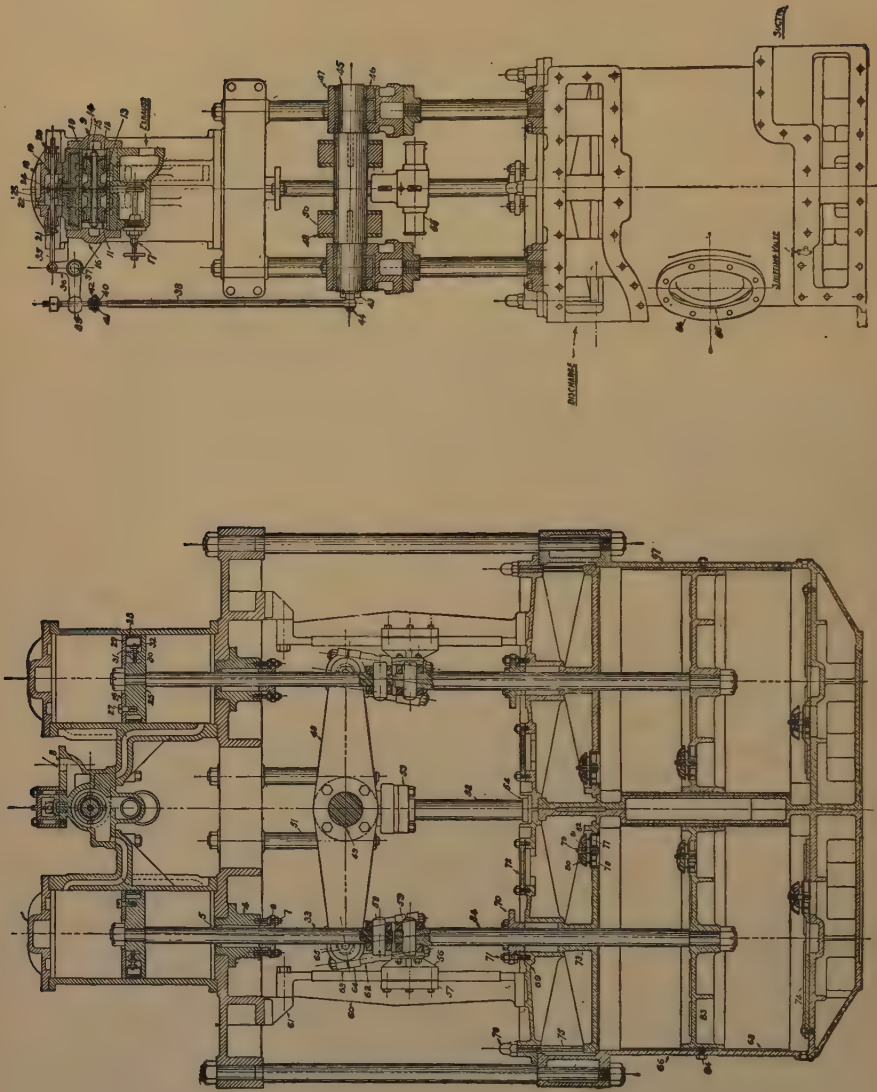


FIG. 81.
Blake & Knowles Vertical Single Acting Twin Beam Air Pumps.

Independent Air Pumps.—In the naval service and in high speed engines in the mercantile marine, the air pumps are not attached to the main engines, but are separate and driven by their auxiliary engines. The advantages of this system are the following:

1. In getting up steam and warming the main engines, the independent pump cares for the condensed steam without waste.

2. In getting underway, and in promptly executing orders from the bridge, the main engines can be handled with a facility which is impossible when they have the air pumps connected directly to themselves.

3. In cruising the independent air pump need only be driven at a speed that will make a good vacuum; and not being subject to

NAMES OF PARTS, FIG. 81.

No.	Name of Part.	No.	Name of Part.
1	Steam cylinder.	45	Beam shaft.
2	Upper steam cylinder head.	46	Beam pedestal.
3	Entablature.	47	Beam pedestal cap.
4	Steam piston rod stuffing box.	48	Beam slab.
5	Steam piston rod box throat bushing.	49	Beam slab key.
6	Steam piston rod gland.	50	Beam slab distance piece.
7	Steam piston rod gland stud and nut.	51	Upper supporting rod.
8	Steam chest.	52	Lower supporting rod.
9	Steam chest back bonnet.	53	Upper supporting flange.
10	Steam chest lining.	54	Lower supporting flange.
11	Steam chest piston valve.	55	Crosshead.
12	Piston valve ring, outer.	56	Crosshead slide.
13	Piston valve ring, inner.	57	Crosshead slide cap.
14	Piston valve follower.	58	Crosshead wedge, steam end.
15	Piston valve bolt complete.	59	Crosshead wedge, pump end.
16	Steam chest front bonnet.	60	Crosshead guide.
17	Exhaust throttle valve complete.	61	Crosshead guide bracket.
18	Auxiliary steam chest.	62	Beam link.
19	Valve rod box throat bushing.	63	Beam link, upper box.
20	Valve rod gland.	64	Beam link, lower box.
21	Valve rod gland stud and nut.	65	Beam link bearing pin complete.
22	Auxiliary steam chest slide valve.	66	Pump cylinder, left hand.
23	Valve rod nut.	67	Pump cylinder, right hand.
24	Valve rod nut taper pin.	68	Pump cylinder lining.
25	Steam piston.	69	Upper pump cylinder head.
26	Steam piston follower.	70	Pump piston rod gland.
27	Steam piston follower bolt.	71	Pump piston rod gland stud and nut.
28	Steam piston ring, outer.	72	Upper pump cylinder head handhole plate.
29	Steam piston ring, inner.	73	Head valve plate.
30	Steam piston set-out bolt.	74	Foot valve plate.
31	Steam piston set-out check nut.	75	Head valve plate holding bolt.
32	Steam piston spring.	76	Head valve plate bolt cap nut.
33	Steam piston rod.	77	Pump valve disc.
34	Pump piston rod.	78	Pump valve seat.
35	Valve rod.	79	Pump valve bolt.
36	Bell crank lever.	80	Pump valve spring guard.
37	Bell crank lever pin complete.	81	Pump valve spring guard nut.
38	Valve rod link.	82	Pump valve spring.
39	Valve rod link tappet.	83	Pump bucket.
40	Valve rod split nut.	84	Pump cylinder lining stud.
41	Valve rod collar.	85	Pump cylinder handhole plug.
42	Valve rod collar set screw.	86	Pump cylinder handhole guard ring.
43	Driving crank.		
44	Driving crank pin complete.		

the wear and tear arising from the "racing" of the main engines, the valves and guards will be maintained in a reliable condition for a long period.

4. When nearing port, and orders to stop are liable to come unexpectedly, the independent pump takes care of the surplus steam which is led into the condenser instead of blowing off from the safety valves. The saving in fresh water thus effected, and also in aiding in care for the boilers, amply repays any small additional expense that may be charged to the operation of the independent pump.

The direct-acting type is applied to all ships constructed in recent years on account of its greater reliability, its simplicity, and the possibility of running at any desired speed without risk of stopping at the end of a stroke.

Independent air pumps are usually made double and operated by the usual steam cylinder and the valve gear of the standard pump makers.

For turbine engines various forms of rotary air pumps have been introduced, as described later.

Air Pump Valves.—Fig. 83 shows the details of the valve, guard and spring. The valves may be made of hard rubber or of sheet metal. The latter is in general use in the navy and gives very excellent results, as they can be made very light, easily cleaned, not affected by grease and have a long endurance. As specified for naval use, they consist each of three circular flat discs of rolled manganese bronze $\frac{1}{32}$ " thick and held in place by a spiral spring with a guard and stud of phosphor bronze, or equivalent material. The valve seat is divided into small openings by ribs, while the lift of the valve is regulated by the guard and securing nut. The valve should be from 4 to 7 inches in diameter, and as light as possible. When metallic valves are used, the three discs are of different diameters, the smaller one being placed on top, and the two lower discs are perforated for water lubrication. Heavier valves have been tried with very unsatisfactory results. The lift of the valve is very important. At high speeds too large a lift prevents the valve from closing quickly and produces pounding; the usual lift given is from $\frac{1}{4}$ to $\frac{5}{8}$ of an inch.

The area of the bucket valves should be as large as possible; as the pumps may have to pass almost solid water, a velocity of about one thousand feet per minute would be the maximum permissible. If the bucket area will not permit of so low a velocity, the stroke should be decreased and diameter increased. Foot valves should have the same area as the bucket valves, while the delivery valves may be increased so as to give as free a discharge as possible.

Another type of air pump valve is shown in Fig. 82.

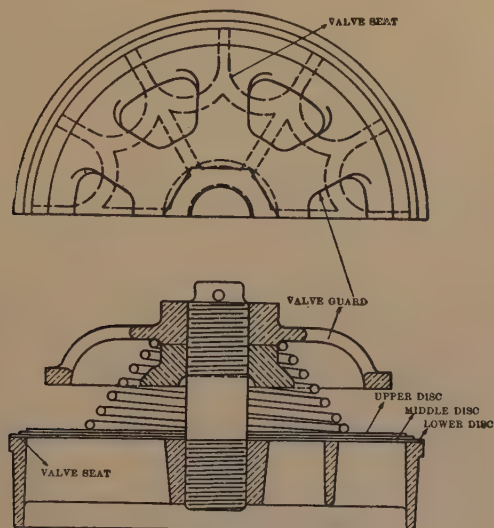


FIG. 82.

Feed Tank and Filter.—The water from the air pump is discharged into the filter and feed tank from which the feed pump takes the water to the boilers. This tank is situated in the engine room and is made of a capacity sufficient for about five minutes full supply for all boilers under full power. The upper part of the tank is divided off for the filter from the lower part by a horizontal partition. Vertical partitions divide this part into three chambers and these are filled with filtering materials, through which the water must pass before entering the lower part of feed tank proper. A vapor pipe, with connections to each filter chamber

and to the lower part of the tank, leads to the atmosphere, carrying off the vapor and contained air in the feed water. There is an overflow pipe leading from the top of the feed tank to the bilge, and arranged so that any water passing down it may be seen. The tanks in each engine room are connected by a pipe between the engine rooms with necessary shut-off valves and each tank is fitted with a water gage. Each feed pump has its suction near the bottom of the tank. The separators, jackets, main and auxiliary steam pipes, radiators and all places where condensed steam can accumulate are fitted with drains, valves and traps which discharge into

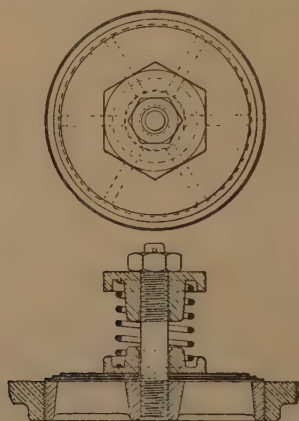


FIG. 83.

the feed tank and enter well below the ordinary water level. A sufficient number of zinc protector plates are suspended from the braces.

Reserve Feed Tanks.—To make up the unavoidable losses of fresh water about the engines and boilers, it is necessary to have reserve tanks from which the “make-up” feed may be taken. The double-bottoms under the engine and fire rooms are usually utilized for this purpose, filled when in port with fresh water from a water boat, or from the *distiller* and *evaporating plant* on the ship. Connections are made to the air pump, to an auxiliary feed pump suction, or to a small double bottom pump; this latter pumping directly from the double bottom to the feed tank.

The most modern air pump systems will now be briefly described, though not used with reciprocating engines.

Dual Air Pump System.—Fig. 84 illustrates a dual air pump system designed to be used in connection with a uniflux condenser.

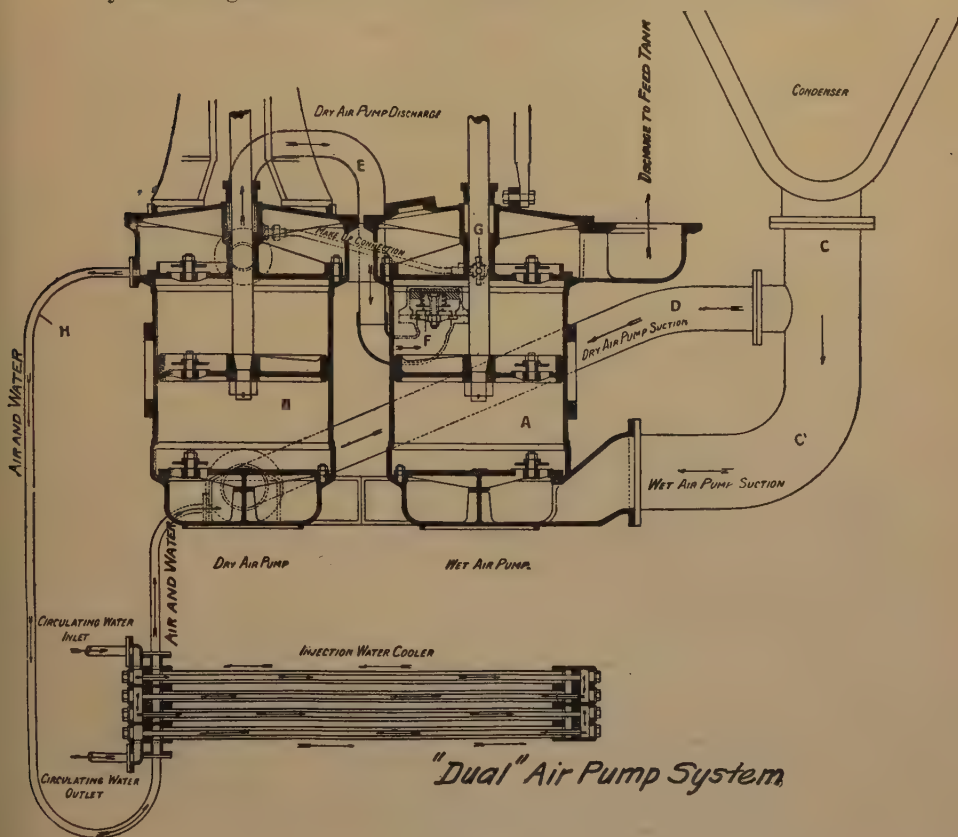


FIG. 84.

It may be used in connection with any condenser, however. This system consists of two vertical pumps worked from one steam cylinder. One of these pumps is called the wet pump and the other the dry pump. In all cases the wet pump is situated below the steam cylinder, as this pump is the only one which works under any considerable load; the dry pump *B* is driven by the beam and

links in the usual manner. One connection *C* is made to the condenser, but a branch pipe *D* is led to the dry pump, the connection being made at such an angle that the *water* will all pass by way of *C'* to the wet pump. Both pumps are generally of the three valve marine type, but in certain cases the dry pump may be of the suction-valveless type. The main features of the dual system are the separate suction for each pump, the dry pump discharging through the return pipe *E*, through a spring loaded valve *F*, into the wet

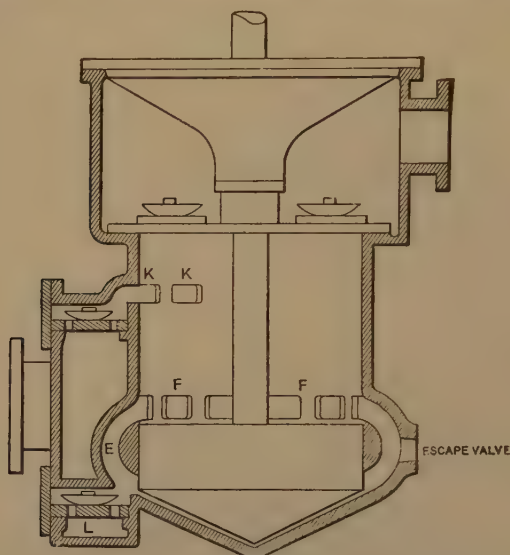


FIG. 85.

pump at a point below its head valves; and the use of the injection water cooler for the dry pump. The supply of water to the dry pump for water sealing, clearance filling, cooling and steam condensing is obtained when starting by opening the make-up valve *G* for a minute or so to enable the vacuum to draw in a supply of water from the hot well of the wet pump. The valve is then closed, and the water passes from the hot well of the dry pump *B* by the pipe *H* to the annular cooler, through which a supply of cold sea water circulates, and after being cooled passes into the suction of the dry pump; then passing through the pump, becomes heated and

again passes to the cooler, and so on in a continuous closed circuit, any excess passing over the pipe *E* to the wet pump.

The spring loaded valve *F* is adjusted to maintain about 20 inches of vacuum in the dry pump hot well when the condenser is working at about 28 inches of vacuum, and this 8 inches difference of pressure is sufficient to cause the water to overcome the cooler friction and pass into the suction, and at the same time never allows any direct air connection between the dry suction and discharge.

Small branch connections from the *main* circulating pump suction and discharge pipes to the injection water cooler of the air pump where shown in the figure, supply the cooling sea water.

A pump of the bucket-valveless type is shown in Fig. 85. The line sketch shows the principle on which it works. On the down stroke a partial vacuum is created above the bucket and upon the latter passing the ports *K* air is drawn into the barrel from the condenser. Meanwhile the air and water which were drawn into the lower part of this pump through *L* during the upstroke, is forced through *E* and *F* into the barrel above the bucket, and the double charge is delivered on the up stroke. The single bucket acts with a double effect. The conical shape of the bucket prevents shock and assists in directing water through passage *E*.

Bi-Therm Air Pump System.—A rotary pump for removing the water, combined with reciprocating air pumps for removal of air, is sometimes known as the “bi-therm” system. This arrangement—not yet installed in our ships—is made possible by using a two stage head and pressure rotary pump which will work under a suction head of only a few inches; the ordinary centrifugal pump requires a greater suction head than is obtainable below the condenser. By connecting the discharge side of the first stage of a two stage pump to a receiver pipe communicating with the vacuum space of the condenser the equivalent necessary head is obtained. The second stage, the pressure pump, withdraws the condensate from the receiver and delivers it to the feed tank. The discharge from the head pump, or first stage, is by a nozzle directed into the suction of the pressure pump, thus utilizing the kinetic energy of the flow.

The water removal pump is entirely independent of the air removal system. The latter may consist of two air pumps, or of twin air pumps, the suctions of which are arranged so as to divide the water from the air as in the dual system. Should the rotary pump fail to handle all the water some will run to the wet air pump suction. Should the rotary pump fail entirely the air pumps will remove the water and still maintain a fair vacuum. The air

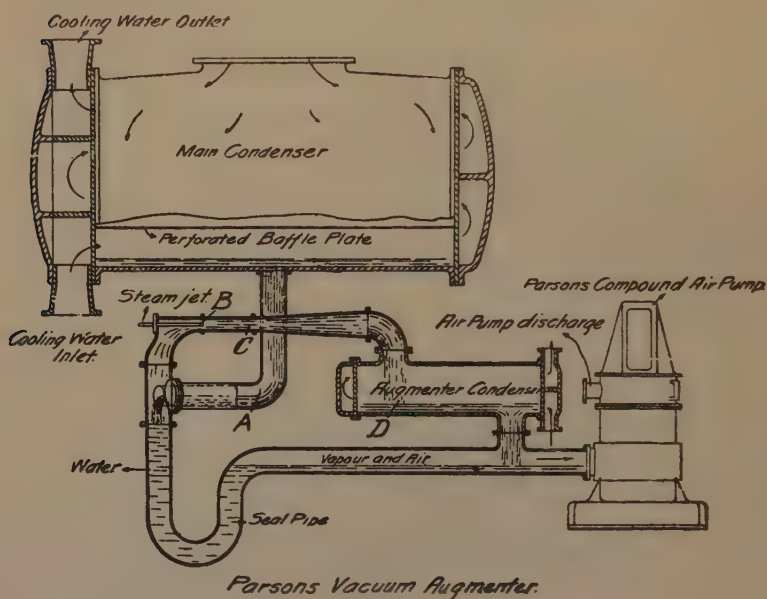


FIG. 86.

pumps are supplied with cooled sealing water, making it possible to speed them up materially to take care of such abnormal air leakage as might occur. In all turbine installations it is desirable that an excess of power for producing vacuum be provided. The importance of this is appreciated upon reflecting that the loss of $\frac{1}{2}$ inch in vacuum in a turbine vessel would reduce the power by nearly 4%. Advantages of the rotary condensate pump are that the water is delivered without shock even though the supply is very

irregular; it occupies very little space; it requires little attention; and it works efficiently regardless of the temperature of the water.

An air pump system to be found in some of our recently built vessels is similar to the "Dual," but has, in place of the injection water cooler, a cooling jacket on the dry air pump suction, and the dry air pump discharges to the atmosphere instead of into the wet pump; an overflow from the dry pump delivery permits water delivered by the dry pump to run to an "overflow tank." The wet pump has a small suction leading to the overflow tank to care for water accumulating there.

Parsons Vacuum Augmenter.—This system has been installed in many of our destroyers. Fig. 86 shows the principle on which it acts.

The air and uncondensed steam are extracted from the main condenser through pipe *A* and are compressed by a steam jet *B* in the contracted pipe *C* to about half their bulk, and are delivered to the air pump at a pressure about 2 inches higher than the pressure in the condenser through an augmenter condenser *D* having 2 to 3 per cent of cooling surface of the main condenser. In this auxiliary condenser the air is cooled and the steam is partially condensed before it enters the air pump. By the extraction of practically all of the air by the steam jet, the conductivity of the cooling surface of the main condenser is much increased. The steam used by the steam jet is about 0.6% of the steam used by the turbine plant at its normal full load. The air pump used is a vertical reciprocating pump of one of the types already described, or a compound pump designed for the purpose.

Kinetic System of Rotary Air Pumps.—Figure 87 shows the general arrangement of this system. 1 is the air suction orifice in the condenser, above the water suction orifice. A vacuum is maintained by a steam ejector 28 (which may use either high pressure, or closed exhaust line steam, as shown), supplemented by a water ejector 4, for removal of air and vapor; water to 4 is supplied by the kinetic pump 6 (a simple centrifugal pump) drawing feed water by way of 7 from the kinetic tank 5 and forcing it through 8 and the water jet 4 back to the tank. The steam in jet 28 and the vapor entrained with air from the condenser are absorbed by the fresh water at 4, the diffuser 9 assisting the air to escape from tank 5 to the atmos-

- | | |
|--|--|
| 1. Kinetic Steam Jet Ejector. | 17. Feed Delivery Pump Suction from Kinetic Tank. |
| 2. Combining Pipe. | 18. Feed Delivery Pump. |
| 3. Non-Return Valve. | 19. Feed Delivery Discharge to Feed Tank or Filters. |
| 4. Kinetic Ejector. | 20. Bypass Valve. To regulate Temperature of 5. |
| 5. Kinetic Tank. | 21. Bypass from Kinetic Tank to Condenser. |
| 6. Kinetic Pump. | 22. Drain Pipe from Main Turbine to Drain Receiver. |
| 7. Kinetic Pump Suction from Tank. | 23. Drain Receiver for Main Turbine Drains. |
| 8. Kinetic Pump Discharge to Kinetic Ejector. | 24. Pressure Water from Kinetic Pump Discharge to Drain Ejector. |
| 9. Air Separating Diffuser. | 25. Drain Water Ejector Nozzles. |
| 10. Air Discharge Pipe. | 26. Drain Ejector Discharge to Drain Pump Suction. |
| 11. Head Pump Suction from Condenser. | 27. Drain Pump for Main Turbine Drains. |
| 12. Head Pump. | 28. Steam Jet. |
| 13. Pressure Pump. } Condensate Pumps. | |
| 14. Stand Pipe. | |
| 15. Pressure Equalizing Pipe. | |
| 16. Non-Return Valve. Pressure Pump Discharge to Tank. | |

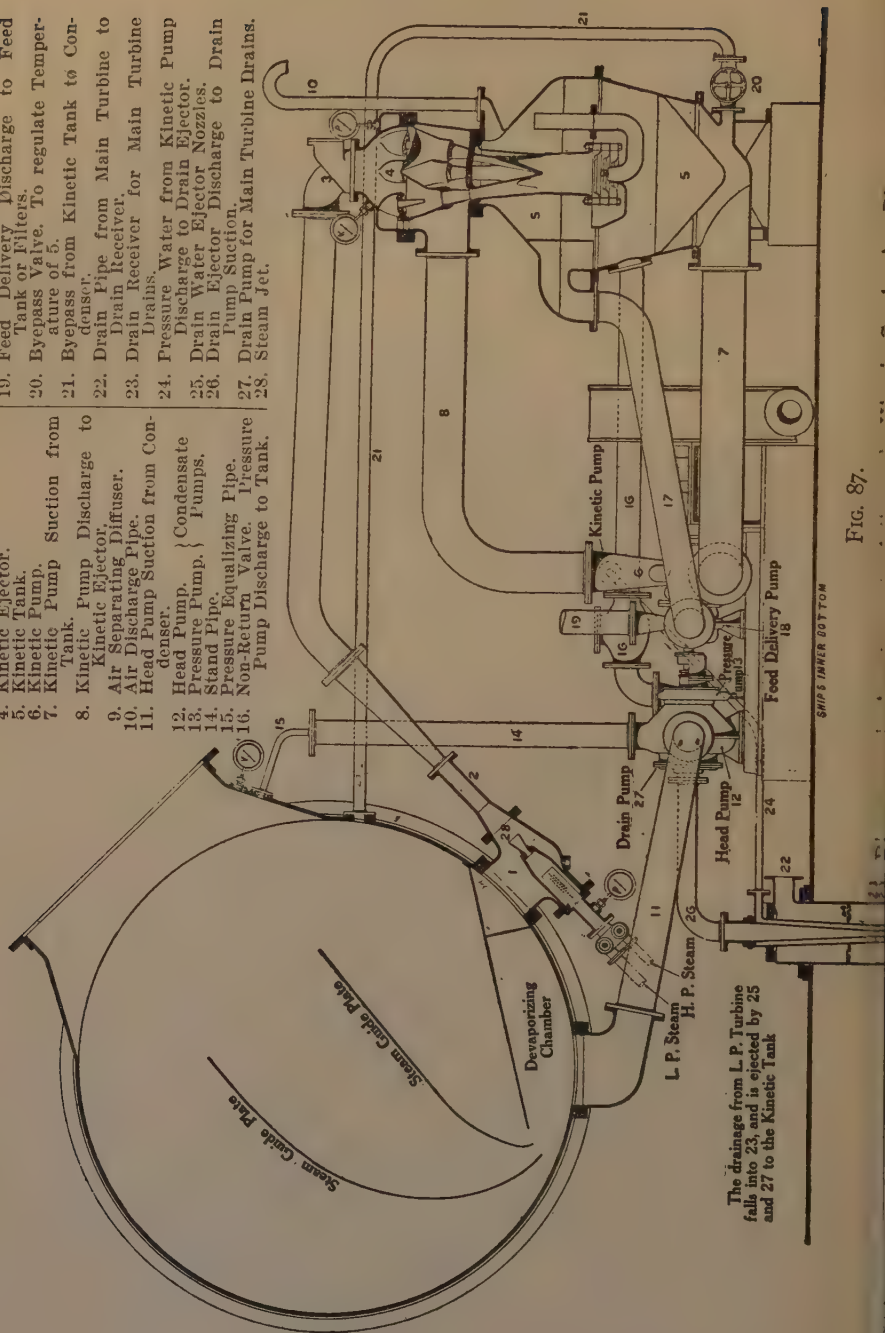


Fig. 87.

phere by way of 10. The construction of 4 is such as to break up the water into small jets, thoroughly mixing it with the steam and air from 2. It can be seen that there is very small loss of heat or water.

The steam water, or condensate, as it is often called, is withdrawn through the wet suction 11 by the head and pressure pumps 12 and 13. The pressure equalizing pipe 15 connects the stand pipe 14 to the vacuum space of the condenser, and, as mentioned under the bi-therm system, this connection enables the rotary pumps to work under a very small suction head. The head pump 12 works under the condenser pressure on both suction and delivery sides; it delivers the water direct to the suction of the pressure pump 13, and the latter delivers through the check valve and pipe 16 to the kinetic tank 5 against atmospheric pressure. The stand pipe 14 provides a natural head of water on the suction side of the pressure pump, and also allows escape of any entrained air.

When the water in the kinetic tank 5 rises the surplus overflows through 17 to the feed tank (not shown).

By opening valve 20 water may be passed to the condenser, and there cooled, thus lowering the temperature of 5 when desired.

The kinetic pump and the head and pressure condensate pumps may be all mounted on one shaft, and driven by a steam turbine or by an electric motor.

It is claimed that the rarefying qualities of the steam and water jets are superior to any combination of reciprocating pumps and that a smaller condenser may therefore be used. The thermal efficiency is high, as the heat in the exhaust steam from the turbine is returned direct to the feed water. Heat lost by radiation is practically the only loss. The temperature of the water used in the jet 4 may be considerably higher than that corresponding to the vacuum in the condenser, permitting a high feed water temperature (see "contraflo kinetic condenser," p. 182).

The Leblanc System.—This system is installed on the fuel ship Neptune, differing in details from the following description.

Figs. 88-91 show the general arrangement of the Leblanc air pumps as applied to the surface condenser. Two rotary pumps are used for removing the air and water. One of these pumps is on the same shaft as the circulating pump. A smaller rotary pump *K*, is used to remove the water from the condenser and has its suction at the bottom of the condenser. The pump for removing air has

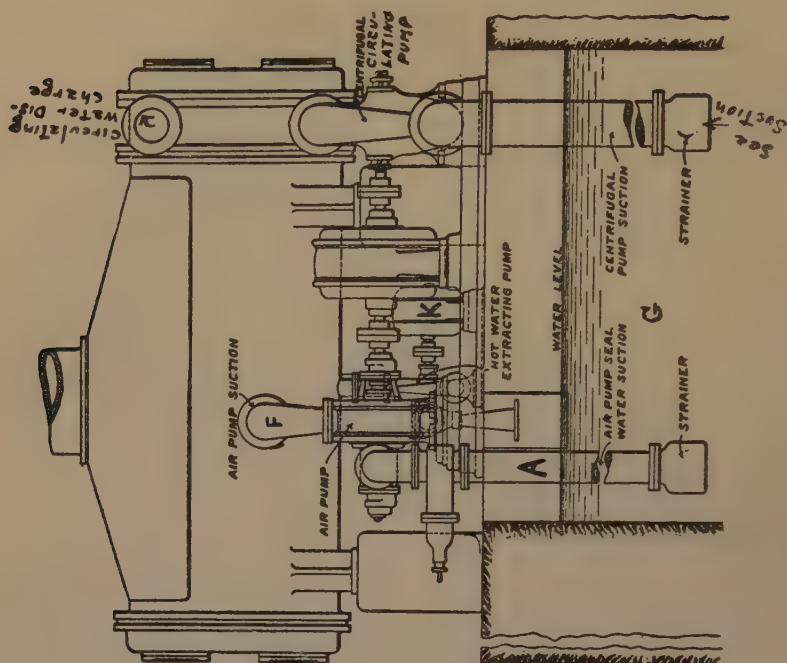


FIG. 88.

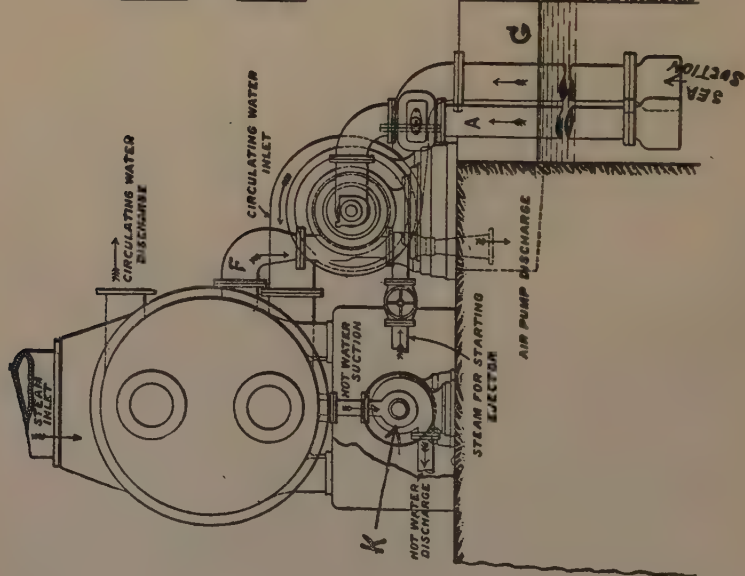
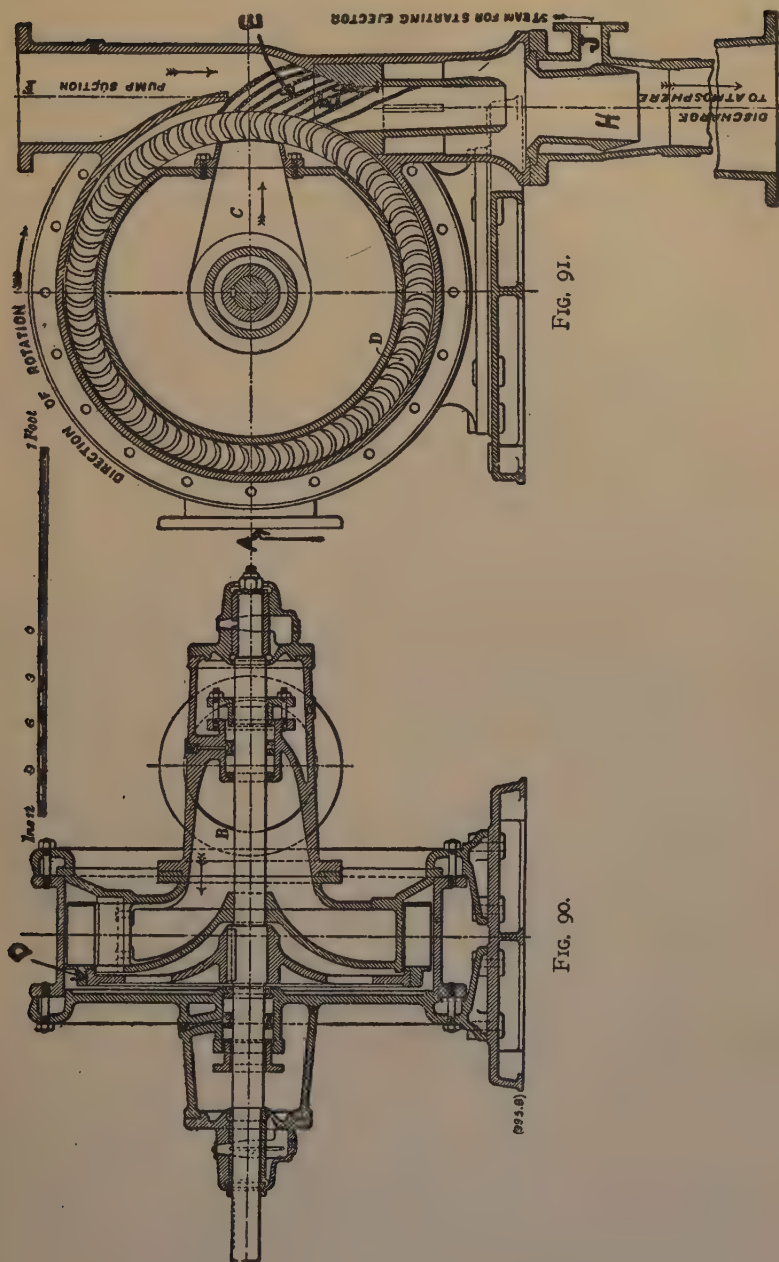


FIG. 89.



a suction in the hot well, and is so arranged as to remove air from the condenser through a suction pipe attached to the condenser above the level of the suction of the water removal pump. The action of the air removal pump is shown in Figs. 90 and 91. Water from the tank *G* (Fig. 88) is drawn up by the air pump and enters the latter by branch *A*, whence it flows into the central chamber *B*, Fig. 90. The only outlet from this chamber is through a horizontal nozzle by way of port *C*, Fig. 91. The impeller of the pump is keyed to the shaft, and its rim is composed of a bronze ring furnished with blades so that the ring and the blades form one piece. The bladed ring *D* runs in an annulus between the walls of the central chamber and the casing. As the blades pass the orifice *C* of the central chamber, they pick up water and project it outwards in thin sheets as indicated at *E*. Air from the condenser, entering by the branch *F*, is entrained by the sheets of water and is discharged with them against the pressure of the atmosphere. The water sheets *E* make an effective seal, so that it is possible for the highest vacuum to be maintained behind it. An ejector nozzle *H* in the discharge pipe, Fig. 91, is for starting the flow of water through the pump. The pump is started and live steam is turned on through the branch *J* at the side of the discharge pipe. A sufficient vacuum is quickly created to enable the pump to pick up its water from tank *G* and commence working. Steam is then turned off of the steam ejector. The sealing water leaves the pump at practically the same temperature at which it enters, as it takes no part in the work of condensation, and it may be used over and over again.

Suction *A* for the air removal pump of course takes water from a fresh water tank, while the circulating pump suction is a sea suction as indicated. A better arrangement than that shown is to have the tank *G* placed higher, so that its water level is above the level of the air pump chamber *B*, allowing free flow of water to the pump and decreasing the necessity for a steam starting-ejector.

As elsewhere stated, for use with reciprocating engines a vacuum of over 26" is unnecessary, and to fully make use of the advantages of higher vacuum necessitates such an increase in the power, weight and size of machinery as to more than offset the direct

advantages of this higher vacuum; therefore, it will be observed that the various air pump systems described above wherein rotary pumps are used are only suitable for turbine installations.

The main features of the principal systems have been described, purposely without much detail; they are undergoing rapid changes in details of construction from day to day with a view to obtaining simplicity, reliability, efficiency, reduced weight and space occupied, and economy of steam consumption.

The Circulating Pump.—The pump in general use for taking the injection water and circulating it through the tubes in the conden-

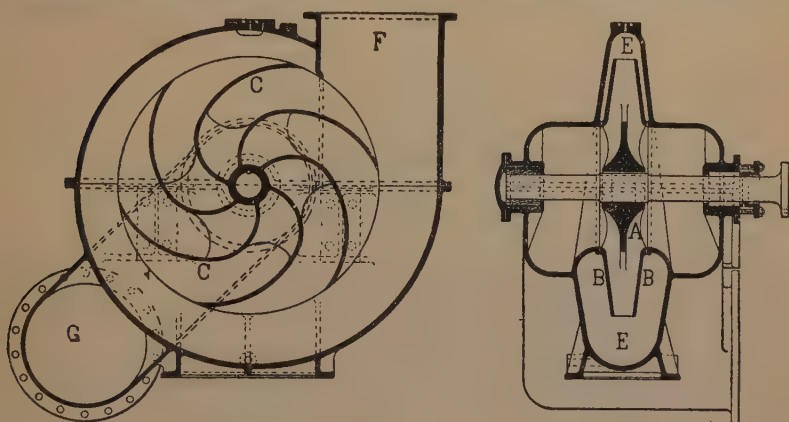


FIG. 92.

ser and overboard is the *centrifugal pump* operated by an independent engine. As the resistance due to the friction of the condenser tubes is but slight, and the quantity of water to be handled is very large, the centrifugal pump is well adapted for the purpose. The independent circulating pump, in addition to those mentioned for the independent air pump, possesses the following advantages over the reciprocating circulating pump, attached to the main engines:

1. The pump can be kept in operation when the engines are stopped, thus keeping the condenser cool.

2. The speed of the pump can be controlled and the amount of the circulating water regulated to suit the occasion.

3. It can be used when the ship is not underway.

4. The large pumping capacity of the centrifugal pump can be utilized in freeing the ship of water in case of emergency.

Centrifugal pumps work very smoothly, have no valves, and if the outboard discharge valve on the condenser should not be

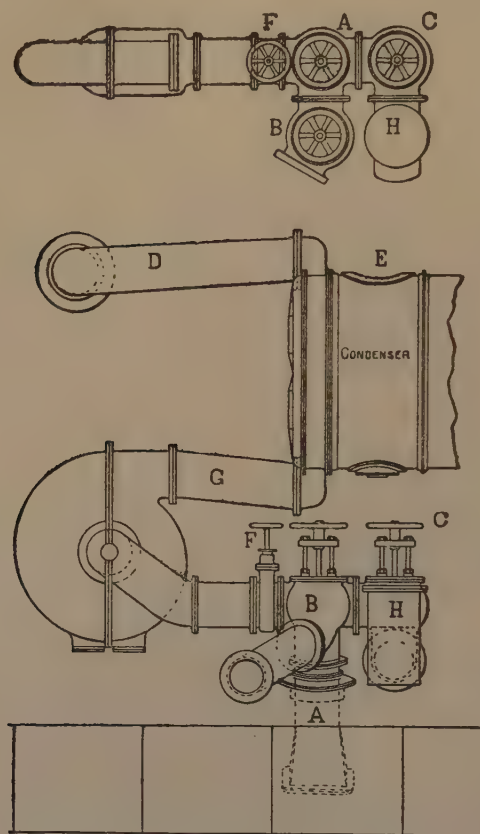


FIG. 93.

opened before starting no injury will result to the condenser, as the water will simply be churned and will bring no pressure of any consequence on the parts.

The details of the centrifugal circulating pump and engine are shown in Fig. 92. The pump consists of an impeller, *A-BB*, com-

posed of a number of vanes, and the casing within which the wheel rotates. The vanes are mounted on a central hub *A*, secured to the shaft with two side plates *B*, inclined towards each other as they approach the circumference, and between which are a series of curved arms *C*, extending from the hub to the circumference. These arms are curved back, away from the direction of rotation, as they proceed from the hub to the circumference. The circumference of the impeller is enclosed by a chamber *E*, in the casing, whose sectional area gradually increases toward the discharge end, from which the delivery pipe *F* is led.

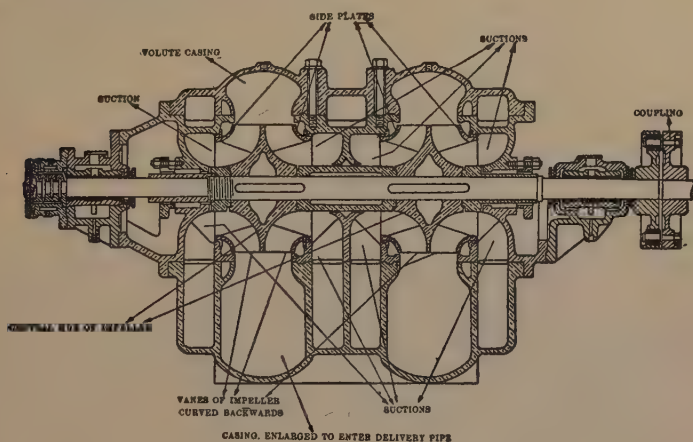


FIG. 94.

The pump suction *G* communicates with the center on each side and the water flowing to the rapidly revolving vanes is thrown outward and around into the casing leading to the delivery pipe *F*, along which it is forced by the centrifugal action of the vanes to the condenser.

The pump casing is made of composition, in two parts so that the runner may be inserted, and the shaft and runner are of phosphor bronze. The shaft bearings consist of lignum vitae strips, dove-tailed in composition split sleeves and lubricated with water.

Though much experimental work has been done on turbine driven circulating pumps they have not yet proven sufficiently economical to supplant the reciprocating-engine driven pump.

Figure 94 shows a recent design of bi-rotor centrifugal circulating pump for a battleship. Two impellers on the same shaft, with suction on each side to balance the thrust; steel shaft encased in brass to protect it from sea-water. Driven by a steam turbine at high-speed. Is a very satisfactory type except as to economy.

Main Circulating Pump Connections.—Each main circulating pump is fitted with suction valves and pipes leading to the sea, to the *main drain*, and to the engine room bilge; to discharge into the condenser through the tubes and overboard, or overboard direct. Each of these suction connections is fitted with a screw-down valve, and the three valve stems are interlocked in such a manner that only one can be opened at a time. Referring to Fig. 93, the circulating pump connection can be seen. *A* is the main injection; *B*, the main drain suction; *C*, bilge suction; *G*, circulating pump delivery; *D*, outboard discharge; *E*, exhaust pipe connection. Between the sea valve and the circulating pump there is always fitted a safety sluice gate valve *F*.

Quantity of Circulating Water Required.—The quantity of heat rejected or dissipated in the condenser depends upon the weight of steam used together with its entering temperature and its temperature after condensation.

Let Q = amount of condensing water required in lbs.

t_1 = temperature of the exhaust steam whose latent heat equals H .

t_2 = temperature of the condensed steam.

t_0 = temperature of the circulating water.

t_3 = temperature of the discharge water.

The total heat to be absorbed by the circulating water per pound of condensed steam is $(t_1 + H) - t_2$.

The total heat taken up by the circulating water is $Q(t_3 - t_0)$.

Hence, $Q(t_3 - t_0) = (t_1 + H) - t_2$, and since $t_1 + H$ = the total heat of evaporation from t_2 and at t_1 ,

$$Q(t_3 - t_0) = 1114^\circ + .3t_1 - t_2.$$

$$Q = \frac{1114 + .3t_1 - t_2}{t_3 - t_0}.$$

Example: Exhaust steam pressure = three pounds absolute or 142° F.

Temperature of water after condensation = 100° F.

Temperature injection = 70°.

Temperature discharge = 100°.

$$Q = \frac{1114 + .3t_1 - t_2}{30} = 35.2$$

That is, 35.2 lbs. circulating water is required for each lb. of steam to be condensed.

Sea Connections and Fittings.—All openings in the hull below the water line for the supply or discharge of water are fitted with valves bolted to nozzles, also called sea-chests, secured to the skin of

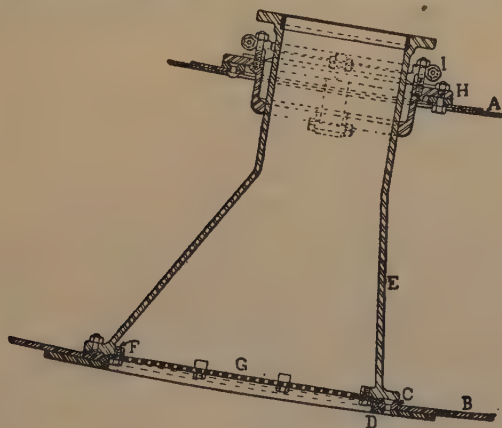


FIG. 95.

the ship. The usual type is of the ordinary screw-down valve for all openings, this having taken the place of Kingston valves through which boilers formerly discharged hot water. The ordinary method of sea connections is shown in Fig. 95, representing the fittings of the main injection valve of a ship having a double bottom. In the outer bottom *B* a hole is cut, around which, to compensate for the loss of strength due to cutting away the material, is riveted a steel ring *C*, in which are fitted a number of stud bolts. A zinc ring *D* is fitted snugly into the hole thus made, and is secured to the ring *C* with counter-sunk screws.

A hole having been cut in the inner bottom *A*, large enough to permit the passage of the bottom flange of the nozzle *E*, this

nozzle is then passed through and secured to the stud bolts in *C*. The nozzle *E* has an inner flange, in which are screwed several stud bolts *F*, whose middle unthreaded portions are square. Over these studs is fitted the grating *G* having square holes to correspond to the square studs, and nuts are screwed on the ends of the studs and secured with split pins run through holes in the studs. A plate *H* made in halves, having a stuffing box cast with it, is then fitted around the nozzle, and is bolted to the inner bottom *A*. Stud bolts are screwed into *H* around the stuffing box, and serve to hold in place the gland *I* (which is also made in halves) and to screw it down after the stuffing box has been filled with several turns of packing.

The zinc ring *D* protects the bottom of the ship from pitting by being eaten away gradually during the electrolytic action which is set up by salt water, the steel skin of the ship, and the composition valve chest.

The grating *G* prevents the entrance of weeds and other foreign matter and can be readily removed, if necessary, when in dock.

The area of the holes in this grating is about twice the area of the valve attached to the nozzle, and this excess area and the large conical mouth of the nozzle permit the inner pipe to receive a full supply of water.

Should the ship ground and the outer bottom be pierced, the water cannot pass the inner bottom, and if the outer bottom be sprung, the nozzle, being free to move in the stuffing box, will not be broken away nor will the inner bottom be affected.

In connection with the text-book on Boilers, the various parts through which the steam passes from the boiler through the engine and back again to the boiler has been taken up in detail. It may be useful to give a brief summary, noting in order the names of the various parts in the circuit, from the formation of steam in the boiler and through a triple-expansion engine.

Boiler to Engine.—Dry pipe, Boiler stop valve, Boiler steam pipe, Main steam pipe, Main and bulkhead stop valves, Separator, Engine stop valve and throttle valve, H. P. valve chest.

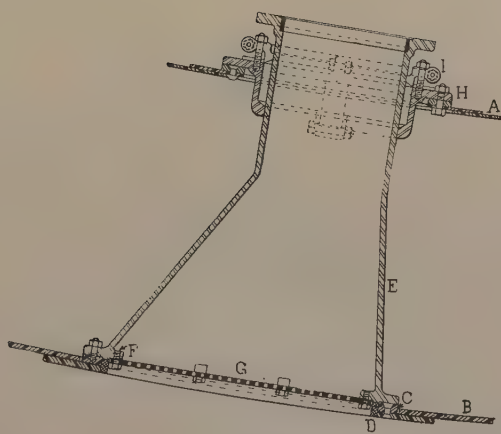


FIG. 95.

H. P. Engine.—H. P. steam ports, cylinder, back through steam ports, exhaust port, 1st receiver, I. P. valve chest.

I. P. Engine.—I. P. steam ports, cylinder, back through steam ports, exhaust port, 2d receiver, L. P. valve chest.

L. P. Cylinder.—L. P. steam port, cylinder, back through steam ports, exhaust port, main exhaust pipe, condenser.

Condenser to Feed Tank.—Condenser, air pump suction; through foot, bucket and delivery valves of air pump; discharge pipe to filter, feed tank.

Feed Tank to Boiler.—Feed pump suction pipe; suction valves, pump barrel, discharge valves; to the main or auxiliary feed pipe, grease extractor, feed heater; feed and check valves, internal feed pipe, and boiler.

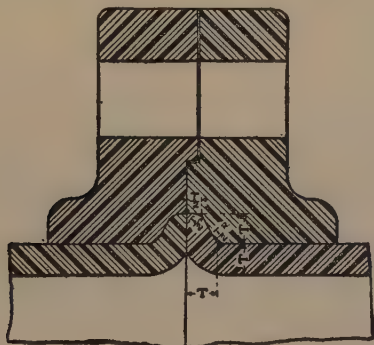


FIG. 96.

Piping.—The piping system in a ship may be divided into three general classes: Steam pipes, exhaust pipes, and water pipes. Steam pipes may be made of copper, wrought iron, or steel. Small and moderate size copper steam piping is solid drawn and in the larger sizes made of sheet copper with brazed joints. Wrought iron pipes are lap welded and steel pipes are made seamless or solid drawn.

The advantages of copper as a material for piping are its ductility, non-corrosive qualities and the ease with which it may be worked into irregular forms as at elbows, etc. It possesses the disadvan-

tages of greater cost, low tensile strength, possible deterioration of the material in working to shape and loss of ductility from service.

The advantages of steel and iron are the reverse of the above disadvantages, lower cost, greater tensile strength, less liability to damage in manufacture and less liability to a loss of strength in service. The disadvantages are the liability to corrosion and greater weight.

In the U. S. Navy the present practice is to make all main and auxiliary steam pipes of steel, seamless drawn, and all steel steam piping is required to be fitted with flanges of steel of the same quality as in the pipe. The flanges are stamped or forged and the pipe is either welded to the flange or else rolled into the flange and beaded over to fit a recess flush with the face of the flange, Fig. 96.

The following table gives the material used in the different classes of piping in the U. S. Navy:

TABLE V.

Pipes.	Material.
All pipes less than 2 inches diameter.	Copper, unless otherwise directed.
All steam pipes 2 inches diameter and above.	Steel, seamless drawn. Copper where directed by Bureau.
Main exhaust pipes.	Sheet steel.
All other exhaust pipes.	Copper, seamless drawn.
All feed suction pipes.	Do.
All feed discharge pipes.	Do.
Pipes to reserve feed-water tanks.	Lap-welded wrought-iron or steel tubing, galvanized.
All air and water pipes of refrigerating plant.	Copper, unless otherwise directed.
Cooling pipes in refrigerating room.	Wrought iron.
Cooling pipes for magazine cooling.	Do.
Cooling pipes in scuttle butts.	Copper, well tinned on the outside.
Suctions from: Bilge, shaft alleys, C. and R. drainage manifolds.	Lap-welded wrought-iron or steel tubing, galvanized.
All internal pipes in boilers.	Brass or steel, No. 14 B. W. G.
All water service pipes on main or auxiliary engines.	Brass, polished above floor.
Oil fuel suction pipes.	Steel, seamless drawn.
Oil fuel service pipes.	Do.
$\frac{3}{4}$ -inch branch pipes to oil fuel burners.	Copper, seamless drawn.
Pipes for forced lubrication.	Copper, or brass, polished above the floor plates.
All fresh-water pipes of evaporating and distilling plants.	Galvanized wrought iron.
Evaporator drains to traps and tanks, and brine and blow pipes.	Brass, iron pipe size.
Distiller circulating pump suction and discharge and evaporator feed pipes.	Copper.
Pneumatic main from air compressor.	Copper, seamless drawn.
Copper and brass pipes 6 inches diameter and less.	Seamless drawn.
Other pipes.	As directed and approved.

The thickness of the piping for use in the U. S. Navy is obtained from the following formulas:

For straight copper pipe $T = \frac{P \times D}{8,000} + \frac{1}{16}$ inch,

For steel steam pipe $T = \frac{P \times D}{10,000} + \frac{1}{16}$ inch,

where P = pressure above atmosphere in pounds per square inch,

D = inside diameter of pipe in inches,

T = thickness of pipe in inches.

For steam, bleeder, blow pipes and suction pipes from the boiler, P = the boiler pressure.

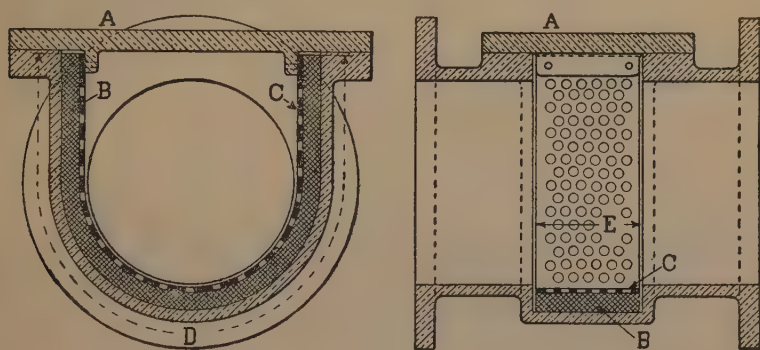


FIG. 97.

For feed discharge pipes, $P = 1.5 \times$ boiler pressure.

The above formulas differ slightly from those in use for piping on U. S. merchant vessels.

All copper pipes for salt water should be protected on the inside by coating of asphaltum or waterproof varnish, and copper piping in the bilges should be well painted and should not rest in contact with the iron or steel work of the ship, care being taken to lead it sufficiently high to keep clear of the bilge water under ordinary conditions.

Zinc Boxes.—Copper suction and discharge pipes designed to convey salt water are fitted with zinc protectors. In Fig. 97 is represented the standard zinc boxes adopted by the Bureau of Steam Engineering. These boxes are of cast composition with flanges to match those on the pipes, and placed at intervals of 10 or 15 feet

on horizontal sections of pipes, the discharge pipe from the pump having one of them placed as close as practicable to the pump. The box has the general form of the body of a gate valve. *A* is the bonnet over the opening for the removal and insertion of the zinc *B*, which is of U shape bent from a rolled slab $\frac{1}{2}$ inch thick. *C* is the guard of thin brass punched with holes of about $\frac{3}{8}$ inch diameter. The length and width of the zinc protectors is given by *D* and *E*.

Joints and Packing.—Joints may be divided into *fixed joints*, as in the case of a joint between the two fixed flanges of a pipe, or between cylinder cover and cylinder flange, and *slip joints*, provided on straight lengths of steam piping to provide for expansion.

In making fixed joints a *gasket* is usually inserted between the two surfaces, and the flanges brought together and made tight by bolts and nuts. The materials used for gaskets exist in great profusion: sheet asbestos, either plain or in combination with other material; rubber, plain or in combination with fibrous material such as cotton or canvas, the cotton and rubber forming alternate layers and called rubber insertion packing; or sheet wire gauze in combination with the above and other materials and known by various trade names. Rubber alone is unfit for steam or hot water joints, owing to its becoming hard and brittle.

For steam pipe joints, asbestos is used; or, red lead, white lead and boiled linseed oil, well mixed into a moderately stiff putty make a good joint; also a joint much used is made of wire gauze covered with red lead. For high pressures a joint is frequently made with soft copper wire inserted in shallow grooves in the faced flanges. On screwing up the flanges the soft copper is flattened and spread out, making a tight joint. For U. S. naval vessels, all joints between flanges of the steam and exhaust pipe are made with corrugated copper gaskets.

All pipes should be led and flanges so placed that they are accessible.

It is generally impossible to make a satisfactory joint on steam pipes unless proper provision has been made for expansion and contraction, and the surfaces of the flanges on the ends of the pipe sections must be parallel to each other.

Expansion or Slip Joint.—The difference in length of a long steam pipe when hot and cold, due to the expansion of the metal, is considerable, and, to prevent an undue strain on the connections, *expansion joints* are fitted at intervals along the length.

Referring to Fig. 98, which shows an expansion joint as fitted on the steam pipes of modern vessels, the parts consist of a composition stuffing box *A*, gland *B*, and entering pipe *C*, the stuffing box and entering pipe having flanges *D* and *E* for connection with pipe main. The stuffing box is packed with metallic packing set up by means of the gland *B* with its studs and nuts, while the enter-

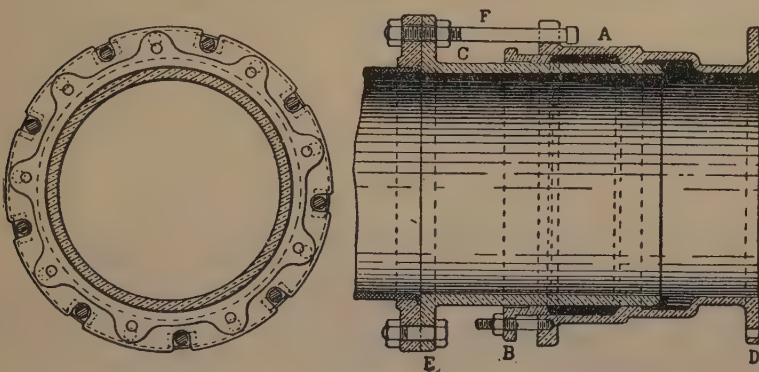


FIG. 98.

ing pipe is free to move in and out for a certain distance and yet remain steam tight. By this means the expansion and contraction of the steam pipe is allowed for, and no strain thrown on the connection.

To prevent this expansion joint from separating, if the anchors, by means of which adjacent sections are held in place, fail to hold, or where two joints occur without an anchor between, stop bolts are fitted for limiting the motion. In the figure, one of the bolts, *F*, is shown at the top. There is a slight play allowed between the stuffing box flanges and the T heads of the anchor bolts to allow for expansion and contraction, while the nuts at the flanges of the entering pipe are set up tight.

When pipes are much curved the curvature permits expansion and contraction, and slip joints are not needed.

CHAPTER X.

THE INDICATOR AND THE TORSIONMETER.

The steam engine indicator is an instrument used in recording, by means of a diagram, the varying steam pressures in a cylinder during a complete revolution of the shaft or double stroke of the piston, and from which the effective pressure at any point of the stroke can be ascertained and the mean effective pressure throughout the stroke calculated.

While the indicator simply gives a graphic measure and record of the pressures in the cylinder, by reasoning and comparison from the indicator diagram many important particulars of the engine, such as the following, may be calculated or inferred:

1. The mean effective pressure of the steam upon the piston, with which can be calculated the *indicated horse power* of the engine.

2. Whether or not the valves are set correctly; the action of the distributing steam valve, whether admission is early or late; the point of cut-off; exhaust opening; closure; compression; and the nature of the expansion curve.

3. The drop of pressure between the boiler and cylinder; whether the full pressure during admission is maintained or the steam wire-drawn, and various other details relative to the action of the steam in the cylinder, and the adjustment and condition of the slide valve and piston.

4. The amount of the back pressure or vacuum, and, in connection with a feed water test showing the actual amount of steam consumed, the economy with which the engine works.

The indicator in its simplest form consists of a small piston working in a cylinder with ample clearance, and carrying a pencil at the end of the piston rod. The upper part of the cylinder is in communication with the atmosphere, while the lower part, by means of a three-way cock and short pipes, may be placed at will in communication with either end of the cylinder or with the atmosphere. A spiral spring is attached to the piston and upper cylinder cover

and regulates the movement of the piston, as the varying steam pressure is brought upon it.

At the beginning of the stroke when the initial steam enters the main engine cylinder, the indicator piston rises to a certain height, depending upon the known elasticity of the spring used. This height remains stationary until cut off, when the pressure within the cylinder commences to diminish, the compression of the spring begins to relax, and the indicator piston gradually falls. To record this action, a rotating barrel or drum is carried in front of the pencil on the indicator piston and actuated by a string attached to the engine crosshead, or some part having a motion coincident with the engine piston, and at right angles to the direction of motion of the indicator piston.

The combination of these two movements produces a rectangular diagram in which the ordinates show, to scale, the varying pressures pushing the piston forward and the resisting pressure to the piston on the backward stroke. The abscissa corresponding to any ordinate shows the proportionate distance moved through by the engine piston.

The objection to the primitive indicator was that the movement of the pencil and indicator piston were the same, but with low steam pressures and slow-running engines this was satisfactory. With high steam pressures and quick-running engines, the long weak springs caused violent oscillations of the pencil and produced a diagram with undulations and wave lines, thus giving very inaccurate and misleading results as to the engine performance.

The springs now used are of high tension, giving a small motion to the indicator piston, and thus reducing the vibration or oscillation. To obtain a suitable height for the diagram, the pencil is carried by a parallel or right line motion, which multiplies the motion of the piston, and still keeps a constant ratio between the motion of the piston and pencil.

The Richard's indicator was the first indicator fitted with a multiplying device for the pencil, and its application was very successful for many years, until the increased speed and pressure and the degree of excellence to which the steam engine had been brought necessitated many material departures from this indicator, espe-

cially in the weight of parts. The modern improvements consist mainly in nearer perfect details of workmanship, reducing the friction to a minimum, lightening the moving parts, using light steel links instead of brass, and reducing the size and height of the drum, with the result that the modern indicator can be relied upon for the speeds liable to be met with in present practice. A few modern indicators that will be met with in daily practice are selected for illustration; it will be seen that they all possess many features in common, their principal difference being in details only.

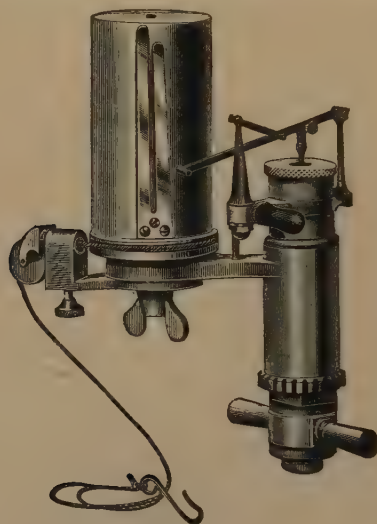


FIG. 99.

The Thompson Indicator.—Figs. 99 and 100 represent the Thompson indicator, and consist of a cylindrical chamber *A*, with an interior shell within which the piston *B* (.798 inches in diameter giving an area of $\frac{1}{2}$ sq. in.) is loosely fitted to work with as little friction as possible. Above the piston, connected to it and to the cover of the cylindrical chamber, is a helical spring of known tension to resist the pressure below the piston, or the atmospheric pressure above the piston when there is a partial vacuum on the under side. The piston rod *C* and pencil *D* are connected to a series of links forming a parallel motion and carried on a movable sleeve around the upper part of the cylinder *A*. By

means of this parallel motion the pencil moves parallel with the piston, but travels through a considerably greater distance, the ratio in the Thompson indicator being 4 to 1. The connection between the links and piston is so arranged that the former can slide freely around the cylinder casing and the pencil easily brought in contact with the recording paper, or withdrawn as desired. The cap of the indicator cylinder has holes in it which admit air freely to the top of the piston.

To prevent the pencil from pressing too hard upon the card an adjustable stop is provided and consists of the rubber handle *E*

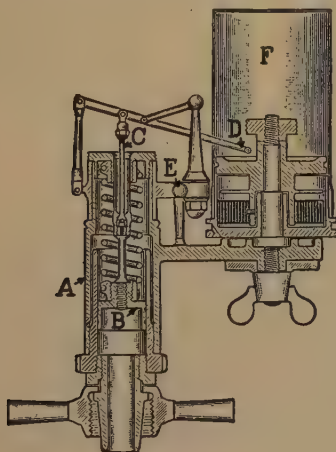


FIG. 100.



FIG. 101.

attached to the sleeve carrying the pencil motion and fitted with an adjustable screw which bears against a small vertical stop or post. The pressure of the pencil against the paper is regulated by screwing this handle in or out. Metallic points are sometimes used instead of pencils, and with chemically prepared paper rolled upon the drum, so that on contact with the pointer a black impression is made.

At the lower end of the indicator cylinder is an ordinary coupling, threaded to fit a three-way cock (Fig. 102) attached to the pipe connecting the ends of the engine cylinder, thus permitting the indicator to be opened to either end as desired, or to the

atmosphere, and enabling diagrams to be taken from each end, on the same card. The holes for the indicator pipes are drilled in the clearance spaces at the end of the cylinders, care being taken in the location that they are not choked in any manner by the motion of the piston.

The indicator card or paper is wrapped around a vertical drum *F* and held in place by clips. The drum revolves on its axis by means of a cord wound around a groove in the base of the drum, and the other end attached to the crosshead or some part of the engine whose motion coincides with that of the piston, while by means of appropriate reducing mechanism the drum is given about three-fourths of a revolution for each stroke of the piston. The string is kept in tension on the return stroke and brought back to

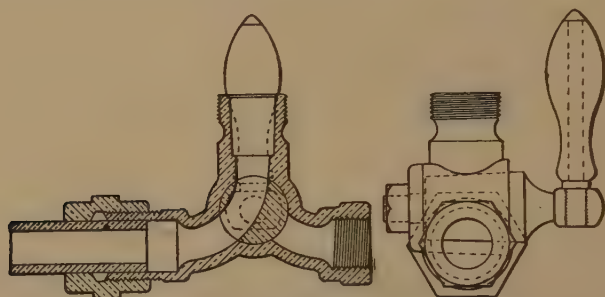


FIG. 102.

its first position by means of a volute or flat spiral steel spring within the drum, thus keeping the string taut, and ensuring coincidence in the motion of the drum and engine piston. The tension of the coiled drum spring can be increased or decreased for different speeds of the engine; as little can be taken up or let out as desired, and thus provide for fine adjustments.

A detent motion enables the drum carriage to be brought to a standstill while connected to the reducing motion, enabling cards to be quickly removed and renewed.

The spring is easily taken out and replaced by another when higher or lower pressures are used, by unscrewing the milled nut at the top of the cylinder, then taking out piston with arms and connections, disconnecting pencil lever and piston, unscrewing

spring from piston, substituting the desired one, and replacing all parts in the same manner.

The Tabor Indicator, whose special feature lies in the means employed in obtaining a straight line movement for the pencil, is shown in Figs. 103 and 104. A stationary plate containing a curved slot is firmly secured in an upright position to the cover of the indicator cylinder, and serves as a guide and controls the motion of the pencil bar. The side of the pencil bar carries a roller, turning on a pin and fitted so as to roll freely in the slot with little lost motion. The curve of the slot is so adjusted, and the pin attached to such a point that the end of the pencil bar which

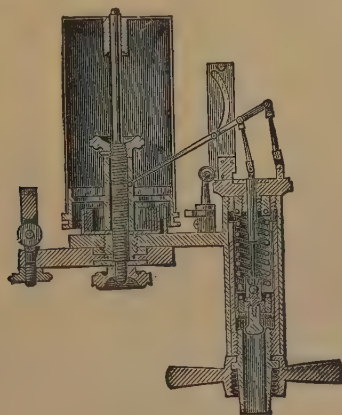


FIG. 103.

carries the pencil moves up and down in a straight line, the curve in the slot just compensating for the tendency of the pencil point to move in the arc of a circle, and a straight line motion results. Inside the cylinder is a movable lining within which the piston works. This lining is attached by means of a screw thread at the bottom, with openings at the opposite sides of the top for the introduction of a tool for screwing in or out when necessary for the renewal of a cylinder.

The slotted plate is backed by another of similar size, which serves to receive the pressure brought to bear on the pencil bar cover, and at the lower end solid with a reduced diameter. The piston is connected to the pencil mechanism by means of a steel

piston rod, hollow at the upper end, where it passes through the cover, and at the lower end solid with a reduced diameter. The connection with the piston is made by means of a ball and socket joint.

The springs are of the duplex type, made of two spiral coils of steel wire strongly held at their ends in brass fittings. This arrangement equalizes side strains on the spring and ensures the piston moving central in the cylinder. The pencil mechanism

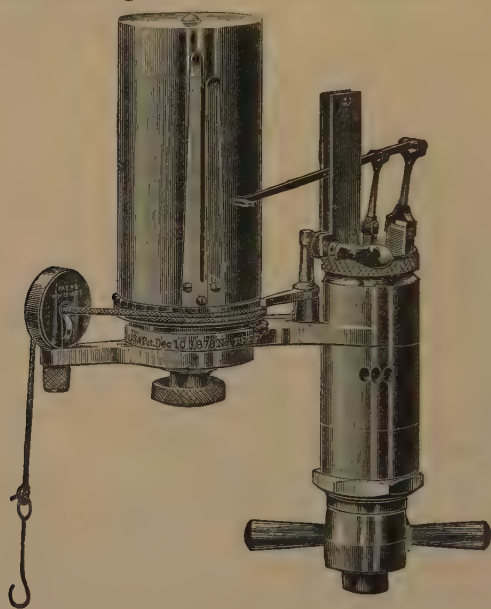


FIG. 104.



FIG. 105.

consists of three pieces: the pencil bar, the back link and piston rod link. The two links are parallel with each other throughout the movement.

This instrument is also fitted with a pencil stop device for regulating the pressure of the pencil on the paper drum. A ratchet is cut on the edge of the drum carriage and a pawl provided, attached to the frame, so as to engage in it whenever it is desired to stop the motion of the drum without unhooking the driving cord.

The pencil point gives a vertical movement exactly five times the travel of the piston.

The Crosby Indicator, Figs. 106 and 107, is another example of a recent type of indicator, and one illustrating the combination of lightness, accuracy and good workmanship, characteristic of modern high-grade instruments.

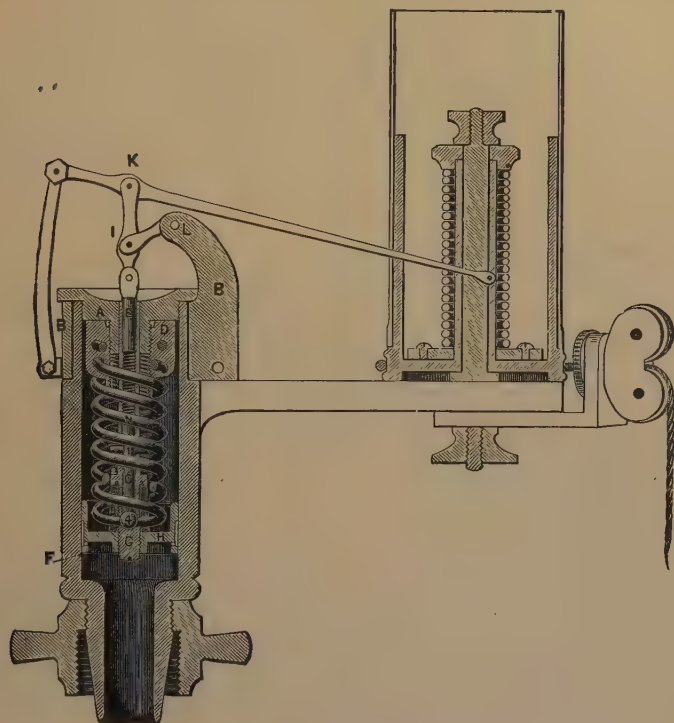


FIG. 106.

A different form of parallel motion from those previously discussed is shown, and is designed for steadiness with the utmost lightness, in order to eliminate as far as possible the effect of momentum in high-speed work. The amount of the movement of the piston and spring is multiplied six times at the pencil point, and to obtain an exact proportional movement of the pencil to the steam pressure on the piston, the spring used (Fig. 108) is a double

helix, wound from one piece of steel wire, the spiral ends being screwed into a brass head with four wings. Instead of a brass foot for the attachment of the bottom of the spring, where lightness is of importance, a small steel bead (+) is attached to the wire with a bearing in the center of the piston, and in connection with the lower end of the piston rod and upper end of the piston screw, it forms a ball-and-socket joint, permitting the spring to yield from pressure in any direction without bending.

The pencil motion is carried by the sleeve *B*, turns freely on the body of the indicator, and is held in place by the cap. The pencil

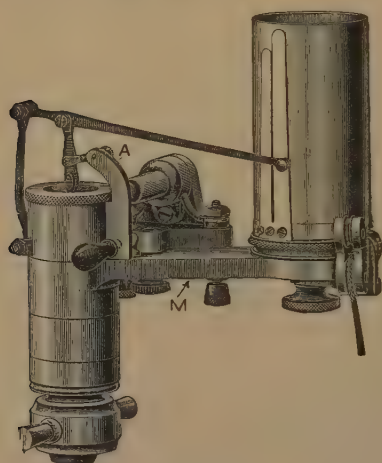


FIG. 107.



FIG. 108.

point is adjusted for light touch on the drum by a handle screwed into the arm and brought to a stop by contact with a stop attached to plate *M*. The piston rod *C* is of steel and made hollow for lightness.

The drum and appurtenances, except the drum spring, are similar in design and functions to other indicators.

If the conditions under which the drum spring operates be considered it will be seen that at the beginning of the stroke, when the cord has all the resistance of the drum and spring to overcome, the latter should offer the least resistance; in the beginning of the stroke in the opposite direction, however, when the spring

has to overcome the inertia and friction of the drum, its energy or recoil should be greatest. A short spiral spring instead of a volute spring is adopted, which fulfils the conditions, giving at the beginning the stroke a slight resistance, increases and reaches

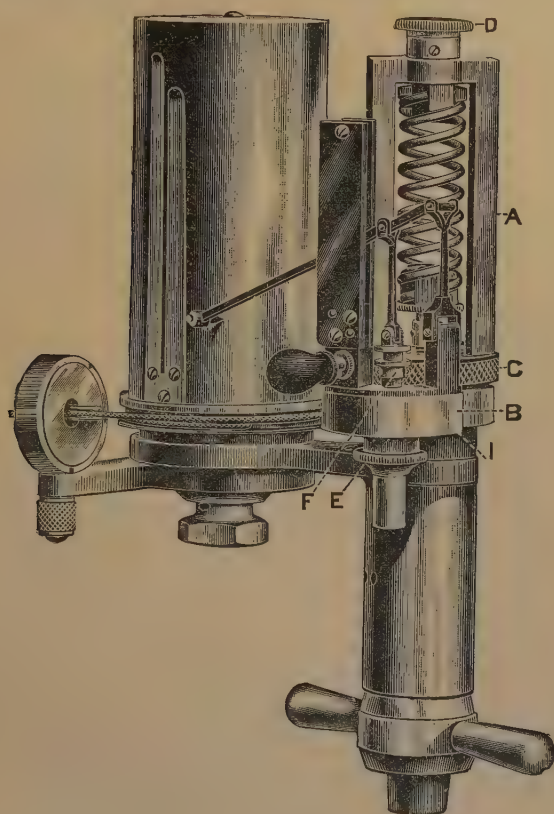


FIG. 109.

a maximum at the end. On the return the recoil is strongest at the beginning of the stroke, gradually decreasing until the end.

The height of the atmospheric line on the drum can be raised or lowered by unscrewing the cap from the cylinder, raising the sleeve carrying the pencil movement, and then turning the cap to the right or left; the piston rod will thus be screwed on or off the swivel *D*, and the position of the atmospheric line raised or lowered.

Outside Connected Spring Indicators.—A modification recently introduced is the indicator with outside connected spring, and is required by the Bureau of Steam Engineering in the specifications for machinery for vessels now under construction.

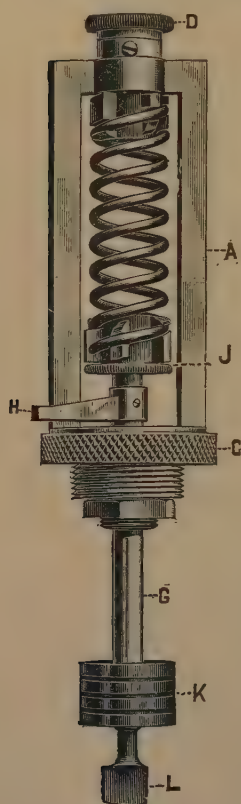


FIG. 110.

Advantages of an Outside Spring.—1. The direct contact of steam with the spring is detrimental to accuracy.

2. The spring being in plain view of the operator, its momentary action can be observed at a glance.

3. The spring is accessible and can be readily removed, without disconnecting any other part.

Fig. 109 shows the *Tabor Indicator* thus fitted. The piston rod has a threaded mounting secured to its top end, outside of the cylinder, to which one end of the spring is secured. The other end of the spring is rigidly held by the thumb screw *D* in the yoke *A*. The yoke *A* swivels in the cylinder cap *C*, which covers the top of the cylinder. The body of the thumb screw contains a circular groove which receives the end of a small set screw for holding it in position, and prevents it from becoming detached when the spring is disconnected.

To disconnect the spring, loosen thumb screw *D*, and unscrew the spring from the mounting on top of the piston rod. A horizontal arm *H*, connects the piston rod to the pencil movement (see Fig. 110). This connection can be instantly made or broken by simply moving arm *H*

to or from the grooved collar *F*, thus allowing the operator to test the freedom of vertical travel of the pencil movement at any time.

Another method of the use of outside springs is that adapted to the *Star Indicator*, shown in Figs. 111 and 112.

The area of the piston has been reduced from one-half of one square inch to one-quarter of one square inch; this allows of the

use of a much smaller size of wire for springs, thereby insuring greater elasticity and freedom of operation.

The spring is elongated or pulled apart instead of being compressed or pushed together as is the practice in other makes of outside spring type of indicators. The claim for the operation of this spring is that when a spring is elongated or pulled apart everything tends to move in a straight line.

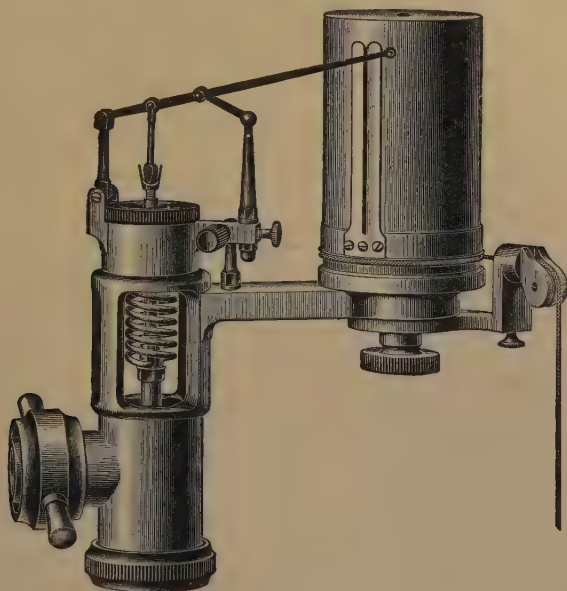


FIG. III.

The operation of pencil movement is directly opposite to that used in the inside spring types, viz., the atmospheric line is at the top of drum instead of at the bottom.

The connection to cylinder is at the side instead of at the bottom as in all other types. This style of connection causes steam to be taken on top of piston.

This indicator is fitted with the Thompson pencil movement, and in the details of cylinder, piston, drum and spring all the requirements of the modern high-grade indicator have been complied with.

Both of the above indicators are fitted to new vessels under construction, and details of their action can be seen in the Laboratory and Model Room.

Preparation and Method of Taking Diagrams.—The indicator should be adjusted and put in proper condition, the joints working freely, the cylinder barrel wiped and oiled, pencil point sharpened and the stop adjusted so that the pencil presses lightly on the

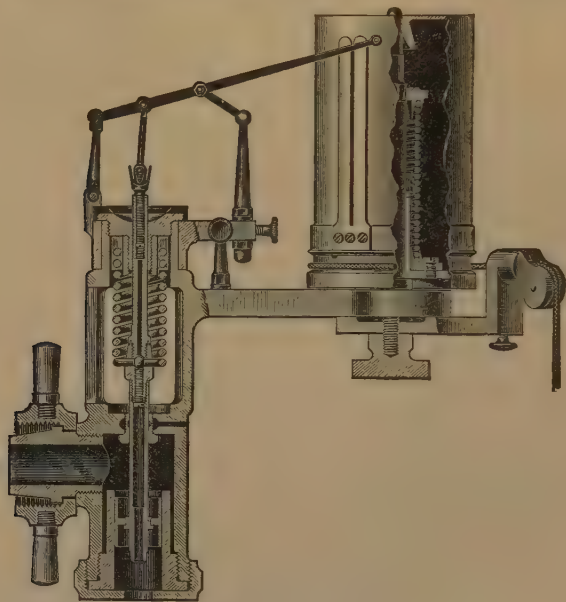


FIG. 112.

paper. Before attaching the instrument, the indicator pipes should be blown through to remove any dirt or sediment that may be present. The indicator is now connected to the indicator cock, the string attached to the reducing mechanism, and its length adjusted by means of attachments usually furnished (Fig. 113), or by any devices by which the length of the string can be readily lengthened or shortened. When adjusted the drum should revolve freely without touching the stops at the ends of its travel.

The card is now placed on the drum and secured by the spring

clips before mentioned, the string hooked on and the drum takes up the motion of the main piston. The indicator cock is now opened to the end of the cylinder from which the diagram is to be taken, and the indicator piston allowed to move up and down until it has become thoroughly warmed. The links containing the pencil are now moved up so as to slightly press against the



FIG. 113.

paper on the drum, when the combined horizontal motion of the drum and the vertical movement of the pencil will cause the latter to trace a closed figure similar to that shown in Fig. 115 and called an *indicator diagram* or *work diagram*. The area rep-

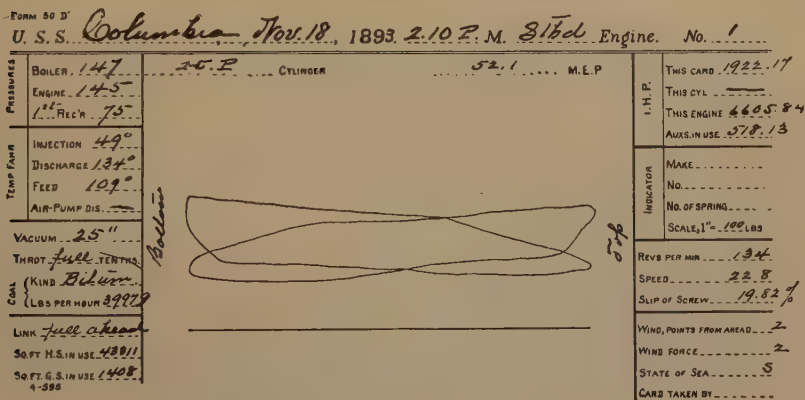


FIG. 114.

resents to scale the effective work done by the steam acting on the piston; the upper part, *EFI*, is drawn on the forward stroke of the engine and shows the varying pressure of the steam on the piston during that stroke, while *JHR* shows the pressure against the piston during the return stroke.

This diagram shows the forward pressure on one side of the piston only, during one complete revolution of the engine, hence

a similar diagram must be taken from the other end by closing the indicator cock to the end under consideration, and opening steam to the indicator from the opposite end, when a diagram similar to the first, but in a reversed condition, will be obtained. As little time as possible should elapse between taking the card at each end. The cock is now closed, shutting off communication with the engine cylinder, and connecting the indicator with the atmosphere, and the atmospheric line is then drawn by the pencil.

Each card as it is removed from the cylinder should have the data filled in as called for on the cards (Fig. 114), the principal ones being time, date, cylinder, end from which card was taken, revolutions, scale of spring, steam and receiver pressure, the registered vacuum, cut-off, etc.

After all the diagrams are taken, the indicator must be removed from the engine (see Naval Instructions, 1913, par. 3054), cleaned, dried, oiled and replaced in its case. The spring should receive especial attention, and care should be taken to prevent any rusting taking place and destroying its accuracy.

Scale of Spring.—Indicators are supplied with springs of 20, 40, 60, 80 and 100 lbs. per sq. in., and the size of the spring employed depends upon the initial pressure used in the cylinder from which the diagram is to be taken, a 20-lb. spring being used on the low-pressure cylinder, with a corresponding higher scale for the others, the scale selected being usually one to bring the diagrams from $1\frac{3}{4}$ to 2 inches in height.

Reducing Motions.—The motion given to the paper drum should coincide, on a reduced scale, with the motion of the piston of the engine. Many arrangements have been devised, and frequently the operator will have to devise new ones to suit the conditions of the case. The pantograph, or "lazy tongs," may be mentioned as a familiar type, the principal objects in the design being to secure one geometrically correct, or one with an error which can be determined and kept within limits.

The Indicator Diagram.—Fig. 115 shows the diagram from one end of a condensing engine; the accepted nomenclature of the different lines composing the card has been briefly mentioned in Chapter II.

The *Atmospheric Line* AB is drawn by the pencil of the indicator when connections with the engine are closed, and both sides of the piston open to the atmosphere.

The *Vacuum Line* OX , used as a reference line; horizontal, and drawn at a distance below the atmospheric line, equal on the scale used to the atmospheric pressure, about 14.7 pounds.

The *Clearance Line* OY is another reference line drawn at a distance from the end of the diagram equal to the same per cent of its length as the clearance volume is of the piston displacement.

The *Line of Boiler Pressure* CD is drawn parallel to the atmospheric line, and at a distance from it equal to the boiler pressure as shown by gage. The difference between CD and EF shows the loss of pressure between the engine and boiler, due to the length of steam pipes, ports and passages in the engine.

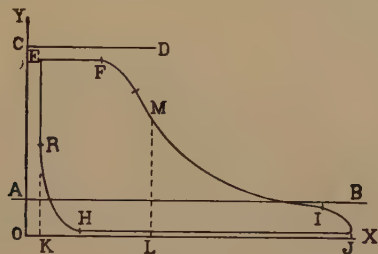


FIG. 115.

Pressures measured from the vacuum line OX are absolute pressures per sq. in.; for instance, at the point L , when the piston has traveled the distance KL , the ordinate LM represents the absolute forward pressure.

The *Admission Line* RE shows the rise of pressure of the steam due to the opening of the valve and the admission of steam to the cylinder. The nature of this line varies with the lead of the valve, nearly vertical with no lead, and inclining outward indicates lead. With no lead or steam late the inclination would be inward.

The *Steam Line* EF is drawn by the pencil when the steam valve is open and steam is entering the cylinder. During the admission of steam to the cylinder, especially in large, quick-working engines, there is a gradual fall of pressure, due to the increased demand for steam as the piston advances, and the steam line,

instead of being horizontal, is sloping. The effect is known as *wire-drawing* the steam.

The Point of Cut-off is the point when the steam or cut-off valve closes, thus stopping the admission of steam, and is usually located at the rounding corner, at that point where the outline changes its curvature from convex to concave. This point depends upon the steam lap of the valve and the angular advance of the eccentric.

The Expansion Curve FI shows the fall in pressure as the steam in the cylinder expands, doing work. The exact form of the expansion curve depends upon the condition of the steam and the manner in which it is used, as discussed in Chapter II. For practical purposes, it is sufficiently accurate when the ordinary hyperbolic curve is used, the absolute pressure varying inversely as the volume.

The Point of Release I shows when the exhaust valve opens.

The Exhaust Line IJ is drawn while the steam is escaping after the exhaust valve commences to open.

The Back Pressure Line JH shows the pressure against the piston during the return stroke, the amount of this pressure being measured from the back pressure line to the zero or vacuum line.

The Point of Exhaust Closure H is the point when the exhaust valve closes, and can be located by the first rise in pressure.

The Compression Curve HR commences from the point of exhaust closure at *H*, and shows the rise in pressure due to the compression of the steam left within the cylinder after the exhaust has closed. The point of closure depends upon the inside lap of the valve and angular advance of the eccentric.

Initial Pressure is the pressure of steam acting on the piston at the beginning of the stroke.

Terminal or Final Pressure is the pressure above the vacuum line at the end of the expansion, assuming the expansion to be carried out to the end of the stroke and the steam not released earlier.

Wire-drawn Steam.—Wire-drawing is the term usually employed to designate the fall of steam pressure due to the frictional resistance of the ports, passages and contracted areas through which it

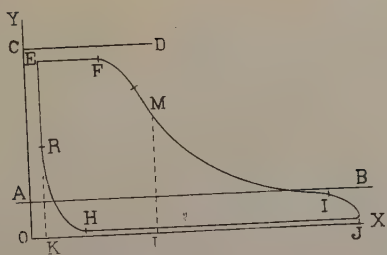


FIG. 115.



passes, and depends upon the speed of piston, condition of the steam and size of ports. Owing to the inertia of the steam and wire-drawing, the steam line of the indicator diagram usually slopes downward. This is further increased with engines at full speed, from the fact that the motion of the piston is not uniform, but gradually increases from the commencement to the middle of the stroke, and the port area being insufficient to admit steam rapidly enough to maintain the full pressure without the employment of very large valves and ports. The slight slope given to the steam line is found of less importance than the reduced size of valves and ports and the inertia of the gear.

The fall of pressure between the boiler and engine varies with the speed, design and length and number of bends in the piping. In well-designed engines of high speed 90 per cent of the absolute boiler pressure should be received at the engine. With long steam pipes, or very wet steam due to insufficient pipe covering and badly-designed valves, the loss is greatly increased. This loss is partially reduced by the drying of the steam to some extent in wire-drawing, or, if the steam is dry it becomes superheated as explained in Chapter II, so that the heat equivalent of some of this lost work is recovered.

During exhaust a similar condition exists, the actual back pressure exceeding the pressure in the condenser. The extent of this difference depends upon the port area and connecting passages, and varies from $1\frac{1}{2}$ pounds at moderate speeds to $2\frac{1}{2}$ pounds at full power. As the valves open and close more or less slowly, the exhaust valve is set to open before the end of the stroke, in order that the vacuum may be nearly complete when the piston commences its return stroke. This early release and gradual opening causes the toe to be rounded as at *I* in Fig. 115.

Clearance.—By the term *piston clearance* is meant the distance between the piston and cylinder cover when the piston is at the end of the stroke. *Clearance volume* is the volume enclosed between the piston at the end of its stroke and the face of the slide valve, and includes the volume of the clearance between the piston and cylinder cover, and the volume of the steam port up to the valve face. Clearance is expressed as a fraction of the whole pis-

ton displacement and varies with the type of the engine, and is greater with small engines than with large ones. In naval vessels with flat slide valves, it varies from 10 to 20 per cent of the piston displacement; with piston valves it varies from 12 to 25 per cent; in recent piston valve designs it is as low as 10 per cent.

The effect of clearance is to increase the consumption of steam without a proportional amount of work, to decrease the ratio of expansion, and to modify nearly all the calculations relating to the steam engine. The steam is wasted on admission, unless there is *compression*, because this clearance volume must be filled with steam each stroke before the piston is moved.

Clearance is usually represented on the diagram by moving the zero line of volume AA' (Fig. 116) back through a distance C equal

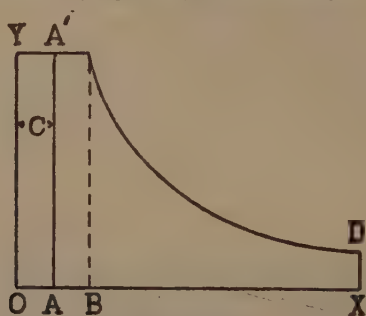


FIG. 116.

to the fractional amount of the stroke represented by the clearance volume. In the diagram AX is the volume swept through by the piston, $C = OA =$ clearance volume, the apparent ratio of expansion,

$$r' = \frac{\text{Final volume}}{\text{Volume at Cut-off}} = \frac{AX}{AB}.$$

The real ratio of expansion = $\frac{OA + AX}{OA + AB} = \frac{C + AX}{C + \frac{AX}{r'}} = \frac{C + 1}{C + \frac{1}{r'}}$

For example, suppose the steam cut-off at $\frac{1}{10}$ with a clearance of $\frac{1}{10}$, then the actual ratio of expansion is

$$r = \frac{\frac{1}{10} + 1}{\frac{1}{10} + \frac{1}{10}} = 5.5$$

So that the actual rate of expansion is but $5\frac{1}{2}$ times instead of 10, as given without considering the effect of clearance.

To Find the Ratio of Clearance from the Diagram.—The amount of clearance, or clearance volume is usually determined by the actual measurement of the spaces not swept through by the piston, but since the expansion and compression curves are nearly that of the rectangular hyperbola, a fairly approximate measure can be found from the diagram, by obtaining points on the clearance line by a method the inverse of that given in Chapter II for the construction of the hyperbolic curve. To find the ratio of the clearance volume to the volume swept through by the piston in one stroke, lay off the vacuum line OX (Fig. 117) parallel to the atmospheric line MN on the diagram. Take any two points, a and b on the compression curve and c and d on the expansion line. Draw horizontal and vertical lines through the points forming the rectangles $aa'bb'$ and $cc'dd'$. A diagonal $a'b'$ or $c'd'$

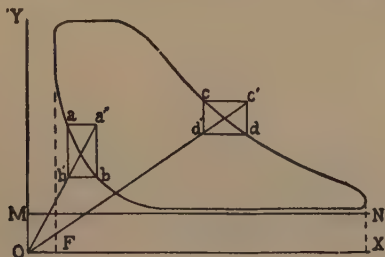


FIG. 117.

drawn through either rectangle and prolonged will intersect the vacuum line at a point in the clearance line OY , and the ratio of clearance is $\frac{OF}{FX}$.

Compression or Cushioning.—During the return stroke of the piston, exhaust closure takes place at some point before the completion of the stroke, as at H (Fig. 115), and the volume of steam left in the cylinder finally occupies the volume of the clearance space. By properly selecting the point of compression, the steam may be compressed to a pressure just equal to that of the admission steam. With this condition, clearance has no effect on the steam consumption nor on the efficiency, for this compressed steam simply forms a cushion which is alternately expanded or compressed with the movement of the piston. If the amount of

compression is less than the above, or absent altogether, the clearance space must be filled each stroke, increasing the consumption which is only partially reduced by the increased area of the diagram. Compression has also the mechanical advantage that it serves to overcome the momentum of the reciprocating parts, and to reduce the strains upon the connections caused by the sudden application of pressure at admission, and furnishes work for the piston while its speed is being reduced.

Compression produces a loss by increased back pressure, but the loss is more than covered by the gain in reducing clearance waste.

Errors Shown by the Diagram.—The theoretical diagram has been illustrated and explained in Chapter II. The lines are parallel and vertical, corners square, and the expansion curve usually the common hyperbola. The actual shape of the diagram and the

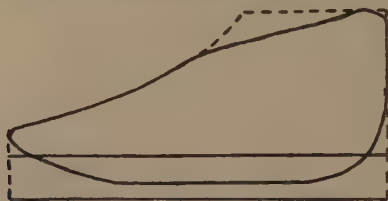


FIG. 118.
CONTRACTED STEAM AND EXHAUST.

deviation from the theoretical indicator card depends upon the manner in which the steam is admitted to and released from the cylinder of the engine; the variety of shapes given from different engines, and from the same engine under different circumstances, being almost endless. The following are a few of the prominent errors or deviations as shown by the application of the indicator:

Small Steam and Exhaust Ports.—This is due to errors in design and construction, and can hardly be remedied except by new parts. Fig. 118 shows the effect on the diagram from such defects. Steam is wire-drawn on steam side with a loss of initial pressure. On the exhaust side, owing to the difficulty of the steam escaping, the back pressure is high, and the vacuum not attained until near the end of the stroke, occasioning a loss of work, and, consequently, area, extending nearly the entire length of the diagram.

Steam Ports and Passages.—Since in most engines the steam has to exhaust through the same ports and passages by which it is admitted, the size of the ports should, as a rule, be governed by the velocity of the exhaust steam rather than that of admission. In general practice, the port area is designed for the exhaust, and the velocities assigned for this now are 5000, 6500 and 8000 feet per minute for the H. P., I. P. and L. P., respectively.

Area of port for high-pressure cylinder =

$$\frac{\text{Area of piston} \times \text{piston speed in feet per min.}}{5000}.$$

For the main steam pipe the area should be calculated so that the mean velocity of flow through it should not exceed 8000 feet per minute. When this is not exceeded the loss of pressure be-

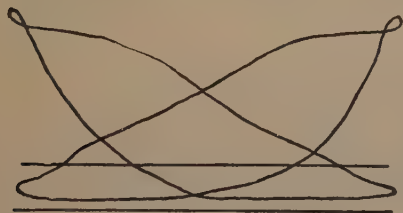


FIG. 119.

EXCESSIVE ANGLE OF ADVANCE.

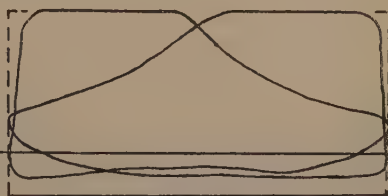


FIG. 120.

ANGLE OF ADVANCE TOO SMALL.

tween the boiler and steam chest is very slight. The area of the exhaust pipe is determined by assuming a mean velocity of flow of exhaust steam of 6000 to 8000 feet per minute.

Excessive Angular Advance.—If the eccentric is set too far in advance of the crank, a diagram similar to Fig. 119 will result. In this diagram all functions on both sides are “too early,” cut-off, exhaust opening and closure, and with steam lead and compression excessive. This effect can be produced in an ordinary engine by “linking in,” as discussed in Chapter V.

Insufficient Angular Advance.—In this case all operations of the valve on both sides will be “too late,” as shown in Fig. 120. The delay in exhaust opening prevents the back pressure line from falling to the condenser pressure until quite a fraction of the stroke has been performed, and thus decreases the area of the dia-

gram and power of the engine. This defect is a more serious one than the one previously described when the operations are too early.

Valve Stem too Short or too Long.—If the valve is incorrectly set on the rod, so that the valve stem is too long (Fig. 121), the result on the diagram, for an outside valve, is as follows: The valve will show too much steam lap at the top and exhaust lap at the bottom, and, consequently, too little steam lap on the bottom and exhaust lap at the top, occasioning:

Top end: Late and small steam opening.

Early cut-off.

Exhaust opening early and full.

Exhaust closure late.

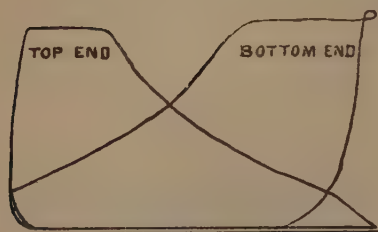


FIG. 121.

VALVE STEM TOO LONG.

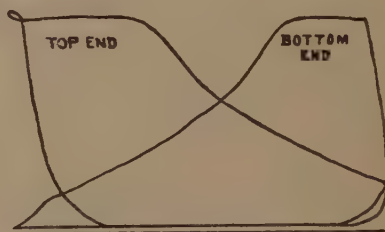


FIG. 122.

VALVE STEM TOO SHORT.

For the bottom end of the cylinder exactly the opposite conditions prevail—steam opening early, and late cut-off; exhaust opening late, and closure early.

If the valve stem is too short, the defects are precisely similar, except that they occur at the opposite ends (Fig. 122).

Excessive Steam Lap.—Lead and steam opening late, early cut-off, port opening small, and steam wire-drawn (Fig. 123).

Insufficient Steam Lap.—Large lead and early steam opening, cut-off late and full port openings (Fig. 124).

With excessive exhaust lap, the exhaust closure is early and compression large, all exhaust openings are late and exhaust lead small. If the exhaust lap is insufficient, the opposite conditions result: the exhaust closure is late, compression small, and exhaust opening early.

If the compression is excessive and carried above the admission pressure, a loop is occasioned at the upper corner as the admission valve opens and the pressure falls. With excessive expansion or when working with no load, or light load, the pressure may fall below that in the next receiver or exhaust space, forming a loop at the end of the diagram as in Fig. 130.

Leaky Slide Valve and Pistons.—A leaky slide valve will permit the entrance of the steam after the valve has cut off, slightly raising the expansion line and causing a loss of efficiency. Similarly, if the piston leaks, steam will escape from the steam to the exhaust side without the performance of work, and proves a direct loss, the expansion line will probably be lowered slightly and the back pressure increased.

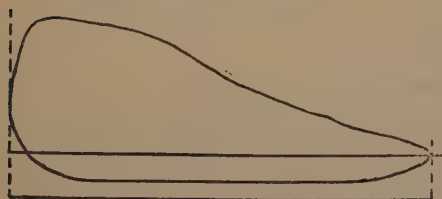


FIG. 123.

EXCESSIVE STEAM LAP.

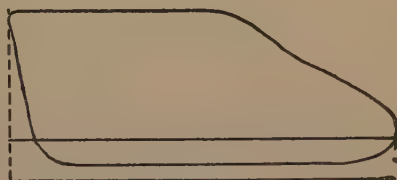


FIG. 124.

INSUFFICIENT STEAM LAP.

In the high or intermediate-pressure cylinders of a stage expansion engine, this loss is counteracted to a great extent, as any steam leaking past the pistons of those cylinders can perform work in the succeeding cylinder.

The leakage of valves and pistons is a matter of considerable importance as regards economy, and is often overlooked in the ordinary working of engines. The indicator diagram is not very useful to detect leakages of this nature. Except in rare instances, or where the leakage is abnormally great, it has so little effect on the lines of the diagram that it is difficult to detect it. The only satisfactory way to determine the tightness of an engine is to apply full boiler pressure to the valves when closed and the engine at rest, then opening the indicator cocks at each end, any leakages can be readily discovered. By blocking the engine with the port closed, steam can be admitted to either side of the piston and any

leakage by the piston can be determined by opening the indicator cock on the opposite side.

Initial Condensation and Re-evaporation.—In order to show this effect on the diagram, it is necessary to draw the hyperbolic curve of expansion. As stated before, this curve does not represent the true expansion of the steam; it is the line to which the expansion in the best types of engines most closely approximates, and is used as a standard of comparison.

In Fig. 125, construct the hyperbola at the point of cut-off, including the clearance as already explained. In this diagram, the expansion curve *CD* of the original diagram lies above the hyperbola throughout its whole length, showing a considerable re-evaporation of water, which has been formed from initial condensation or from water carried over from the boilers, and the difference in



FIG. 125.

the areas shows the work equivalent of the heat given up by the cylinder walls during expansion.

It is usual in determining the amount of cylinder condensation to draw the hyperbolic curve from a point corresponding to the terminal pressure upward to the horizontal line corresponding to the boiler pressure; the difference in the areas of the two diagrams shows the loss of cylinder condensation.

Steam Pipe and Steam Chest Diagrams.—The fluctuations of pressure in the steam pipe and in the steam chest, especially in an automatic cut-off engine, can be ascertained by attaching an indicator to the pipe and to the chest, and if the initial pressure is abnormally below what might be expected, the part causing throttling might be located. These fluctuations may be due to insufficient area of steam pipe or sharp bends, a badly designed stop valve giving insufficient opening, or tortuous or contracted passages to the cylinder.

In Fig. 126 is shown a steam chest card transferred to its proper location on the engine diagram, and such as would be given by an engine fitted with an automatic expansion gear. At *A* the top end of the cylinder starts to receive steam and the pressure falls to *B*; from *B* to *C* during admission the pressure falls slightly as shown by *BC*. At *C* the cut-off closes, causing a sudden rise of pressure to *D* and from this point to the end of the stroke the line is horizontal or slightly inclined downward until at the end of the stroke the steam valve opens, the pressure falls, and the operation is repeated.

Mean Effective Pressure.—The mean forward pressure during a stroke is the mean height of the figure, *ADEFX* (Fig. 127), measured from the vacuum line or line of no pressure, or a mean

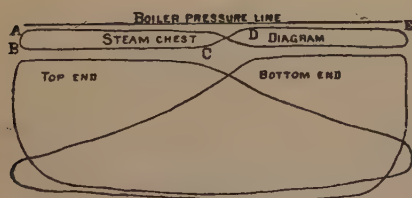


FIG. 126.

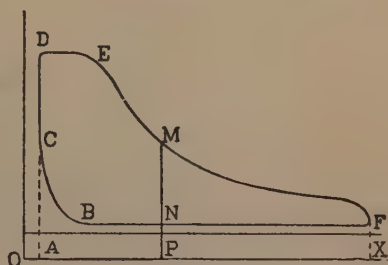


FIG. 127.

of all the ordinates such as *PM*. The mean *back* pressure is the mean height of the figure *ACBFX*, or of the ordinates such as *PN*, measured from the zero line of pressure. The difference between these two mean heights or distances is equal to the effective mean pressure on the side of the piston under consideration during two strokes. To obtain the power of the engine it is necessary to repeat this operation for the other end of the cylinder, and obtain the effective mean pressure for that end, and the mean of these two pressures will be the total M. E. P. on the piston.

It must be understood that to obtain the true effective pressure at any point of the stroke, such as *P*, it is necessary that diagrams be taken from both ends of the cylinder, for though theoretically the two diagrams should be nearly alike, practically they differ materially, and the effective pressure is the difference between the

forward pressure on one side and the back pressure on the other side. Referring to Fig. 128, showing the diagrams from both ends, at the point *P* the forward pressure is *PN*, and at the same time there is a counter pressure or back pressure from the other end, and resisting the piston, equal to *PM*, so that the resultant or effective pressure at this point is *MN*. At the point *C*, when the expansion curve cuts the compression curve, the forward pressure equals the back pressure, and from this point to the end of the stroke the back pressure exceeds the forward pressure, and the effective pressure is negative and tends to bring the piston to rest at the end of the stroke. In the figure, the combined diagram of effort on the piston is shown by the shaded portions *ABCM* and *CDE*, the former being positive and the latter negative. It is thus

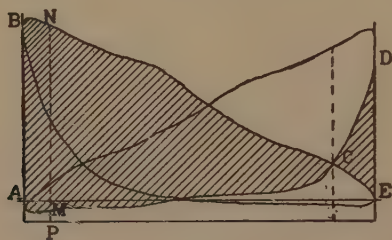


FIG. 128.

seen that to obtain the effort on the piston at any given point, it is necessary to combine the simultaneous diagrams taken from the opposite ends of the cylinder. The correction of the indicator diagrams and the inertia of the moving parts of the engine are considered in the Diagram of Crank Effort, in the following chapter.

Computing the Indicated Horse Power from the Diagram.—As previously explained, the mean effective pressure is the difference between the mean height of the ordinates of the curve of forward pressure *DEF*, Fig. 127, and the mean height of the ordinates of the curve of back pressure *FBC*, both measured from the vacuum line *AX*. Let *S* = scale of indicator spring.

$$\text{Mean forward pressure} = \frac{ADEFXA}{AX} \times S \text{ in lbs. per sq. in.}$$

Mean back pressure = $\frac{AXFBCA}{AX} \times S$ in lbs. per sq. in.

Mean effective pressure = $\left(\frac{ADEFXA}{AX} - \frac{AXFBCA}{AX} \right) S$
 $= \frac{BCDEFB}{AX} \times S,$

or the mean effective pressure equals the area of the diagram *BCDEFB*, divided by the length *AX* and multiplied by the scale of the spring.

The usual method of finding the mean length of the ordinates between the forward and back pressure lines is to divide the atmospheric line or the vacuum line between the limits of the card into ten equal spaces, and lines are drawn across the diagram perpendicular to the atmospheric line and passing through the points

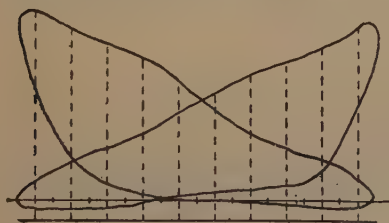


FIG. 129.

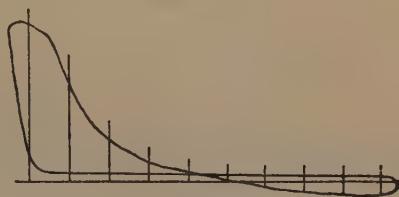


FIG. 130.

midway between the points of division as in Fig. 129. Measure the length of the ordinates between the upper and lower lines of the diagram for each end separately, by a scale of inches or by the indicator scale corresponding to the spring used. The mean of the ten measurements gives the mean effective pressure in pounds per square inch, if measured by the indicator scale; or, if measured in inches, it is converted into the corresponding measurement of pounds pressure per square inch.

In the case of a non-condensing engine under light load the expansion line often extends below the back pressure line and forms a loop in the diagram, as in Fig. 130. The ordinates below the back pressure line are negative, and the sum of their lengths must be subtracted from the sum of the lengths of the ordinates above the back pressure line, in obtaining the M. E. P. In measuring the area of such a card with the planimeter, the instrument

will, of itself, subtract the area of the small loop, if the tracing point be moved over the expansion line as the pencil moved when the loop was formed.

Let P_e = effective pressure in lbs. per sq. in., A = area of piston in sq. in., L = length of stroke in feet, and N = number of revolutions, or double strokes per minute.

Then $P_e LAN$ = foot pounds of work done per minute by one end.

Since the common acceptance of a horse power is 33,000 foot pounds per minute, the indicated horse power developed by one end of an engine is $\frac{P_e LAN}{33,000}$.

At the crank pin end the area of the piston is reduced by the area of the piston rod, while the mean effective pressure also varies, hence the indicated horse power for both ends is given by $\frac{(P_e^c A^c + P_e^h A^h) LN}{33,000}$.

Where $P_e^c A^c$ and $P_e^h A^h$ are the pressure and area at the crank and head ends, respectively.

To simplify the work, the area of the piston at the crank end and head end are so nearly equal that the error made by substituting for each of them $\frac{1}{2}(A^c + A^h)$ is hardly appreciable, and is done to arrive at the common formula for indicated horse power,

$$\text{I. H. P.} = \frac{2PLAN}{33,000}.$$

In this equation the only variables are P and N , and to facilitate calculations the remaining quantities for a given engine are combined to form the constant, $C = \frac{2LA}{33,000}$.

$$\text{I. H. P.} = C \times P \times N.$$

The indicated horse power consists of the power exerted by the steam, and is made up of the net power delivered and, in addition, that consumed in propelling the engine itself.

Dynamometric Horse Power—Prony Brake.—The power of the engine as *delivered* from the crank shaft is the actual or available horse power. Measurements of this latter power are made by either absorbing or transmitting dynamometers, the most common

of the former type being the *Prony Brake*, while of the latter the Emerson Dynamometer is one type, and these can be seen in the machine shops.

There are many forms of the Prony Brake, among the simplest is that illustrated in Fig. 131, and in use in the Laboratory. The actual horse power is measured by opposing external frictional resistance to the engine equal to the power exerted. The apparatus consists of a circular friction brake, consisting of a flexible iron strap lined with wooden blocks *A*, and encircling a pulley on the shaft. A long arm *B* is connected to the strap and is pre-

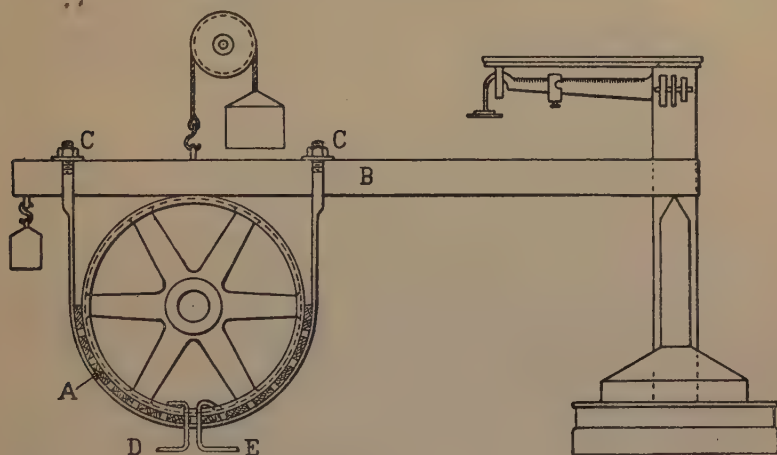


FIG. 131.

vented from turning by having its end in contact with a platform scale, and at the same time the force produced by the friction between the band and pulley is transmitted to, and measured on the scales. When the engine is started the nuts *CC* are gradually screwed up so as to press the wooden blocks hard against the pulley and thus regulate the tension of the band and alter the friction.

The frictional resistance should be adjusted to let the engine run at its usual working speed and the revolutions carefully counted. The line joining the point of action of the weight and the top of the pulley must be kept practically level. Owing to the heat developed a stream of water is used during the test,

usually by providing the rim of the brake with a deep internal flange forming a trough. Water enters the trough through the pipe *D*, revolves with the pulley, and flows out through the scoop or waste pipe *E*.

Instead of using the beam scale, the tendency of the arm to rotate can be balanced by weights hung in a scale pan at the end of the arm when the direction of rotation of the shaft tends to move the arm upward.

To find the *brake* or *shaft horse power*, i. e., the work of the shaft: The action of the brake is equivalent to that of a pulley with a radius equal to the distance between the center of the shaft and the end of the lever exerting the pressure, and is lifting a weight equal to that shown by the scale.

Let P = pressure exerted by the end of lever arm.

L = distance from center of shaft to point of contact on scale.

N = revolutions per minute.

$$\text{Then, Brake Horse Power} = \frac{2\pi PLN}{33,000}.$$

The only work to be put upon the machine during the trial is that of the brake.

If the indicated horse power is taken at the same time as the dynamometrical, the difference is the friction horse power, or the horse power expended in driving the engine against its own friction.

The mechanical efficiency of the engine or machine is the ratio of the dynamometrical or brake horse power to the indicated horse power and varies from 80 to 90 per cent.

$$E = \frac{\text{B. H. P.}}{\text{I. H. P.}}.$$

Planimeters.—In the calculation of mean effective pressures from a diagram it is customary to use the planimeter, whereby time and labor are saved in the computation on a large number of diagrams, and greater accuracy obtained. One form of the planimeter is the *Coffin Averaging Instrument*, illustrated in Fig. 132. This planimeter has one end of the tracing arm guided in a straight grooved metal plate connected to the board on which

the instrument is mounted. The indicator diagram to be measured is placed under the clamps *C* and *K* so that the atmospheric line is near to, and parallel with, the lower edge of the stationary clamp *C*, while the extreme left hand of the diagram is even with the perpendicular edge of the clamp. The movable clamp *K* is then brought to the other end of the diagram. The beam of the instrument is placed on the board with the pin at the lower end resting

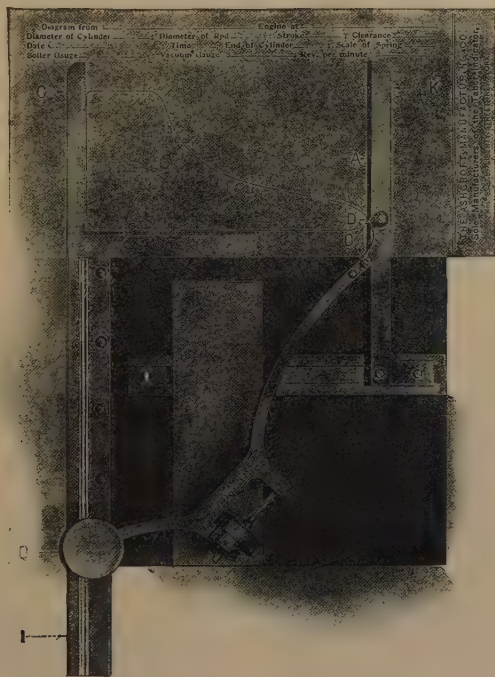


FIG. 132.

in the groove, the weight *Q* applied to the top of the pin to keep it in place, the tracing point moved to the right-hand end of the diagram and set at the point *D* on the line of the diagram, and a light indentation made. The graduated wheel is set so as to bring the zero mark to the zero mark on the vernier. The tracer *O* is now moved over the line of the diagram in the direction of the motion of the hands of a watch until a complete circuit is made, and the tracer finally brought back to the point *D*; the

tracer is now moved upward along the movable clip till the reading on the wheel returns to zero. Another light indentation, *A*, is made in the paper to mark this new position of the tracer. The instrument is now removed and the distance between the points *D* and *A* is equal to the mean height of the diagram and can be measured by the proper scale.

This instrument determines the desired result without computation. The area of the diagram may also be found. This area is found by the reading on the graduated wheel when the circuit of the diagram has been made and the tracer reaches the starting point *D*. This area divided by the length of the diagram gives the mean height.

The wheel has fifteen main divisions, each of which represents one square inch of area. Each division has five subdivisions, each subdivision representing one-fifth of a square inch. The vernier scale enables the subdivisions to be read to fiftieths, each of these fiftieths representing two one-hundredths of an inch.

Steam or Water Consumption from the Indicator Diagram.—From the indicator diagram calculations are frequently made of the steam or water consumption of an engine as *accounted for by the indicator*. The calculated results are always considerably less than the quantity of steam used, and do not give a good idea of the actual steam consumption, since there is likely to be a considerable amount of water in the cylinder present with the steam, due to cylinder condensation and radiation and waste from other sources. This application of the diagram is of great importance, however, when comparing the economy of similar engines, or of the same engine when working under different conditions, and in conjunction with a feed water test.

The indicator diagram shows at any instant the pressure of the steam and the volume occupied, while from a steam table the weight of a cubic foot of steam at the given pressure may be obtained. To find the steam accounted for by the indicator per stroke, select any point from that portion of the diagram when the ports are closed; that is between cut-off and release, or on the compression curve, after exhaust closes. Find the volume in cubic feet of the piston displacement to the point selected, including

clearance, and multiply it by the weight per cubic foot corresponding to the pressure at the point considered on the diagram. This result will be the steam used as shown by the indicator, and when multiplied by the number of strokes per hour and divided by the indicated horse power, is the steam accounted for per I. H. P. per hour.

If the engine is working with compression, the steam found at the terminal pressure is not all exhausted, but a portion of it filling the clearance space is retained, the weight of which at the pressure of compression should be deducted from the above.

Let a = piston area in sq. in.

n = number of strokes per min.

c = clearance in percentage of stroke.

l = length of stroke in feet.

l_1 = length of stroke in feet to the point to be measured.

l_2 = length of return stroke uncompleted at compression.

w = weight of cubic foot of steam at pressure of the point to be measured.

w_1 = weight of cubic foot of steam at pressure of compression.

The weight of the steam selected for consideration is

$$\frac{a}{144} (l_1 + cl) w \text{ pounds per stroke.}$$

The steam restored by compression is

$$\frac{a}{144} (l_2 + cl) w_1 \text{ pounds per stroke.}$$

The net weight used per stroke

$$= \frac{a}{144} [(l_1 + cl) w - (l_2 + cl) w_1]$$

Weight of steam used per I. H. P. per hour =

$$\frac{60n \times \frac{a}{144} [(l_1 + cl) w - (l_2 + cl) w_1]}{\frac{P_e l a n}{33,000}} =$$

$$\frac{13750}{P_e} \left[\left(\frac{l_1 + cl}{l} \right) w - \left(\frac{l_2 + cl}{l} \right) w_1 \right]$$

Example.—The cards (Fig. 133) from a compound engine tested by Mr. Barrus have the following data: *

Cylinders	13" x 26" x 18" stroke
Revolutions per minute	195
Vacuum	27"
Initial pressure absolute	115 pounds
Weight of feed water per hour	3970 pounds

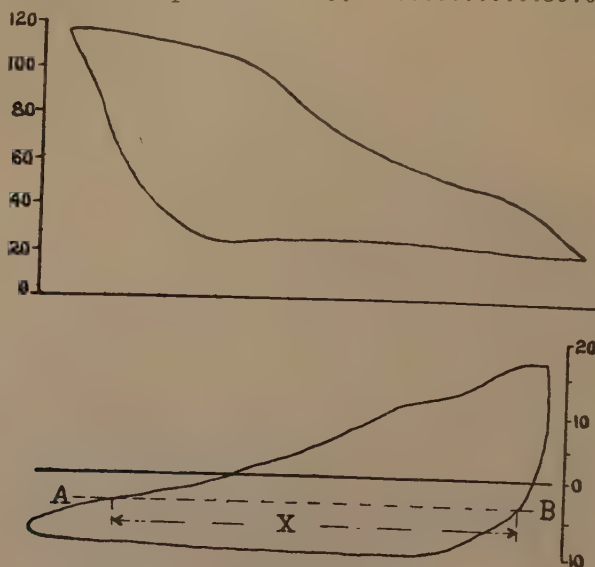


FIG. 133.

	H. P.	L. P.
Clearance	7%	10%
Ratio of cylinders	1	4
Indicated Horse Power	—	—
Total Indicated Horse Power	—	—
Feed per I. H. P. per hour	—	—

Find the I. H. P. of each cylinder, the steam accounted for by the indicator per I. H. P. per hour at cut-off and release in both the H. P. and L. P. cylinders. Also, the proportion of water accounted for at release and cut-off.

* "Engine Tests" by Geo. H. Barrus.

Taking the L. P. cylinder, the M. E. P. referred to this cylinder is 23.76 pounds. The steam accounted for at cut-off is

$$\frac{13750}{23.76} \left[\left(\frac{.558 + .15}{1.5} \right) \times .051 - \left(\frac{.33 + .15}{1.5} \right) \times .0098 \right] \\ = 578 \times (.024 - .003) = 12.14 \text{ pounds per I. H. P. per hour.}$$

Similarly, the consumption per diagram can be obtained at release or any other part of the stroke.

The above is for a simple engine. In multiple-expansion engines the M. E. P. considered is the sum of the mean effective pressures of all the cylinders of the engine referred to the cylinder on which the computation is being made, and is known as the equivalent or *referred* M. E. P. In a triple expansion engine, if the high-pressure cylinder is under consideration, the M. E. P. of the I. P. and L. P. cylinder referred to the H. P. is the M. E. P. of each multiplied by the ratio of $\frac{\text{I. P.}}{\text{H. P.}}$ and $\frac{\text{L. P.}}{\text{H. P.}}$, respectively, and the M. E. P. to be used is the sum of the original H. P. and referred I. P. and L. P. mean pressures. Similarly, if the L. P. cylinder is under consideration the sum of the mean pressures referred to the L. P. cylinder is $\frac{p'_m}{R} + \frac{p''_m}{R_1} + p'''_m$, when R is the ratio of the volumes of the L. P. to H. P. cylinder; R_1 ratio of L. P. to I. P., and p'_m and p''_m are the mean effective pressures of the high and intermediate-pressure cylinders, respectively.

The steam consumption as calculated from the indicator card does not show the actual weight of steam used per horse power by the engine, but simply the amount of steam in the cylinder for the different points of the stroke under consideration. The amount of steam present as shown by the indicator is generally less at cut-off than at release, since some of the water from initial condensation is re-evaporated during expansion.

The preceding method can be much simplified if the position for measurement is selected at some point on the expansion line at or before the release, so that a line through this point and parallel to the atmospheric line will cut the compression curve. This method is much used with actual cards and the procedure is as follows:

Draw a line AB (Fig. 133) parallel to the atmospheric line, above the points of release and compression, below the point of admission and cutting both the expansion and compression lines as shown. Let x equal the length in inches of the intercept of the line AB by the curves, A = the area of the card in sq. in., w = specific volume corresponding to the pressure p at the line AB , s = scale of the indicator spring, l = length of the card in inches, v = volume of the piston displacement in cubic feet, and n = number of revolutions per minute.

$$\text{Water per hour} = \frac{x}{l} \times \frac{v}{w} \times 60 \times n \quad (1)$$

$$\text{I. H. P. of the cylinder} = \frac{\frac{A}{l} \times s \times 144v \times n}{33,000} \quad (2)$$

Dividing (1) by (2),

$$\text{Steam per I. H. P. per hour} = \frac{13750 \times x}{s A w}.$$

The above gives the result for a simple engine. If the calculation is made from one cylinder of a multiple expansion engine the result should be multiplied by

$$\frac{\text{I. H. P. of cylinder}}{\text{I. H. P. of engine.}}$$

Calculate the steam consumption by the above method for the cards shown in Fig. 133.

Combination of Indicator Diagrams in Multiple Expansion Engines.—As mentioned in Chapter VI, there are certain losses in the simple engine which compounding corrects to a great extent but in turn introduces other losses, such as those between the cylinders, consisting of condensation, wire-drawing, friction in the pipes, passages and ports, and the free expansion of the steam between the cylinders without the performance of work. By combining the separate diagrams from the cylinders of a stage expansion engine an approximate idea can be obtained of these losses, and means taken to effect a possible reduction. The combined diagram also furnishes a ready means of making an approximate comparison of the performance of a stage expansion engine with

what would be theoretically obtained if the steam expanded in a single cylinder.

As the scales of pressure and volume of the diagrams from the cylinders of a stage expansion engine are all different, it is useful for the purposes mentioned to combine the diagrams into this single one with common volume and pressure scales, and thus show the continuous relation between the pressure and volume as the steam passes through the cylinders. To do this correctly the clearance in each cylinder should be known and added to the piston displacement, so that the relative lengths of the separate diagrams when combined is as the ratio of the total volume of the cylinders, that is the piston displacement plus the clearance at one end.

The common practice in combining diagrams is as follows: In Fig. 134 draw the lines OX and OY at right angles to each other, the former to represent the axis of the volume and the latter the axis of pressure. Decide upon the scale to be used and lay off from the line OY the lines OA , OB and OC at a distance from the zero line of volume the equivalent of the cubic feet of clearance found at one end of the high pressure, intermediate and low-pressure cylinders, respectively.

Utilize the line OX as the vacuum line or line of no pressure of the combined diagram, and lay off on each of the cards to be combined (Fig. 136) the position of the vacuum line on that card. The indicator cards are then each divided into any number of equal parts, say ten, and ordinates drawn through the middle points.

Starting with the low-pressure card, the piston displacement CG to the scale determined upon is laid off from the clearance line CF . The distance CG is now divided into the same number of equal parts that the original diagram was divided into, and corresponding ordinates drawn as before. Transfer to the scale selected on the respective division lines of the new diagram, the absolute forward and back pressures given at the corresponding division of the original low-pressure diagram. Joining the points thus found by a free curve, will give an elongated diagram from the low-pressure cylinder.

The intermediate and high-pressure diagrams are treated in precisely the same manner, the transferred intermediate diagram is laid off from BD and the high pressure from AE ; the lengths BH and AK being laid off to scale, to represent the displacement of the intermediate and high-pressure pistons, respectively.

Having reproduced the diagrams to a uniform scale, the result is the combined diagram, showing, in a general manner, the expansion of the steam on its way through the different cylinders. For comparison, a hyperbolic curve PS is usually drawn through the estimated point of cut-off, and shows the expansion of the same steam according to the general law of $pv = \text{constant}$.

The difference between the area of the combined cards and that enclosed by the curve PS and the clearance line shows the imperfection of the cycle and the ratio between the two is a fraction known as the *card factor* or *design factor*, which is used in the design of new engines for approximating the expected work with a given low-pressure cylinder, initial pressure and ratio of expansion.

In Fig. 134 a single curve is drawn for the three diagrams, but the amount of clearance is not the same and the quantity of steam taking part in the expansion in each cylinder varies proportionately because of the different amounts of cushion steam in the several cylinders. Consequently, a single hyperbolic or saturation curve cannot correctly represent the curve of the three diagrams and a separate curve should be drawn for each cylinder through its point of cut-off.

The preceding method of combining diagrams may be modified so that the cushion steam may be eliminated and a converted diagram drawn, in which is shown the pressure and volume of the steam actually used or discharged from each cylinder per stroke. A saturation curve drawn under these conditions will be the same for all the diagrams and answer for the complete expansion curve. This is done by deducting from the volume of steam at each pressure, the volume of the cushion steam at that pressure. To obtain this draw saturation curves for the cushion steam in the respective cylinders at the point where compression begins, in each, and then draw a new diagram commencing at the zero line

sure and volume of the constant quantity of steam used. For instance, the transformed diagram of Fig. 136 is shown in Fig. 135, and drawn so that for any point, for instance, M , lay off $O'M'$ from the zero line of volume equal to OS , and make $M'F'$ equal to MF and $M'K'$ equal to MK .

In Fig. 137 is the combined diagram drawn in this manner, from the separate diagrams, Fig. 136. A single saturation curve can now be drawn for a volume equivalent to the cylinder feed per stroke, viz., that quantity supplied from the boiler each stroke. The horizontal distance at any pressure between the expansion curve and the saturation curve DE represents the volume of water present due to cylinder condensation, for instance, if a horizontal

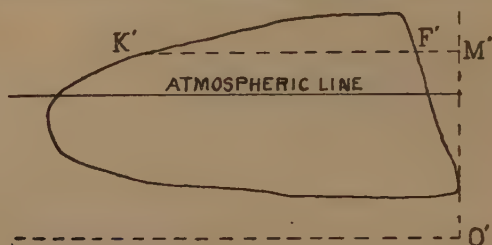


FIG. 135.

line be drawn, AB represents the volume of steam present in the cylinder, and BC the water present.

Except in accurate experimental tests the cylinder feed is not usually known, but if a saturation curve be drawn just touching the transformed diagram, the relation of the diagram to this curve will show the relative condensation as the steam passes through the cylinders.

Standardization of the Indicator.—So much reliance must be placed upon the accuracy of the indicator diagram, it is essential that in all cases of any magnitude or importance, the indicators used should be calibrated, so that the necessary corrections may be applied to the diagram or the principal sources of error eliminated before use. Since the indicator is intended to record the pressure in the cylinder at every point of the stroke, the abscissae of the diagram must be directly proportional to the path of the

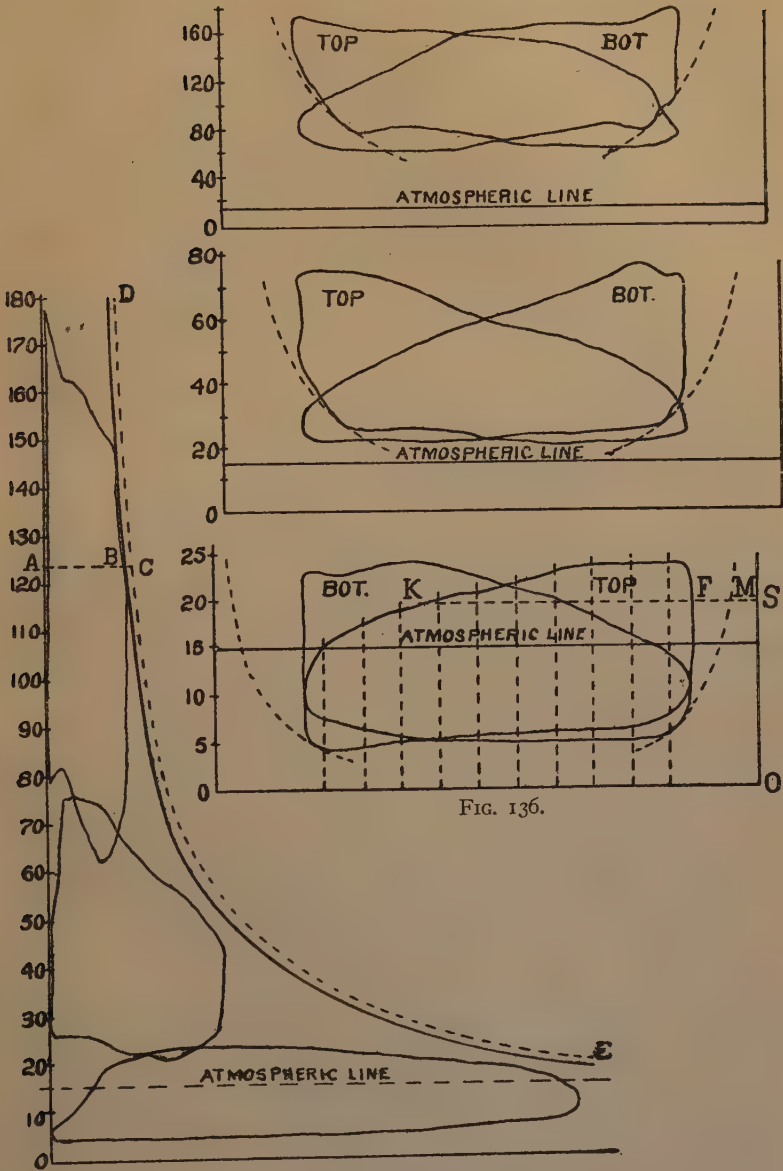


FIG. 136.

FIG. 137.

piston and the ordinates directly proportional to the pressures within the cylinder. These conditions are difficult, if not impossible, to fulfill, and the variations are not the same for instruments of different construction, hence the necessity for calibration before any important investigation, and as nearly as possible under the conditions of actual use.

The following are the principal tests employed:

The calibration of the indicator spring.

The accuracy of the drum motion.

Errors affecting the pencil motion and pencil friction.

Back lash and friction.

The following is the method of comparing and testing indicators and the apparatus used in standardizing indicator springs as adopted by the Navy Department, from an article, "Indicator Tests," by Passed Asst. Eng'r. F. H. Conant, U. S. N., in the "Journal of the American Society of Naval Engineers."

A manifold in the shape of a cylinder, 5 feet long and 3 inches in diameter with cocks, *B*, Fig. 2, Plate 26, is used for attaching the indicators. Steam is admitted through the valve *C*, and released through valve *D*. Communication between the manifold and mercury column is through the water pot, *E*, in which the water is kept at a constant level. The height of the mercury is shown by the index *G*, which is attached to a float in the column. As the scale is graduated for only 25 pounds, the index is shifted on the cord the length of the scale for every increment or decrement of that amount, and the gage *H* is used as an aid in following the index. The manifold and water pot are carefully lagged to prevent condensation, and the column and mercury reservoir are enclosed in a funnel through which air circulates, and keeps all parts of the column at the same temperature.

The indicator cords are led to the pulleys *I*, on the shaft *J*, and a movement of the shaft produces a corresponding movement of the drum. The arrangement for applying the pencils is shown in Fig. 1. A bracket *a*, is secured by set screws to the indicator frame between the cylinder and drum. An arm on the front of the bracket carries a lever pivoted at *b*. The lower end of the lever is connected by the links *c*, with the arm *d*, on the shaft *K*.

The upper end of the lever is connected by the link *e*, with the clamp *f*, set in the handle bracket of the indicator. Attached to the other side of the clamp is the spring *g*, which draws the pencil to the drum. The tension of the spring is adjusted by the screw *h*.

To operate the mechanism, the handle *L*, on the pencil shaft *K*, is raised and draws the pencil about one-sixteenth of an inch away from the drums, where they are held by a fixed detent on the armature of the magnet *M*. The handle *N*, on the drum shaft *J*, is then raised and pulls the drums around about one-third of a turn, where they are held by a detent engaging with the sector *O*. The machine is now set. To trip it, the push button *P*, is pressed, the magnet releases the pencil shaft, and the pencils are drawn to the drums. The arm *Q*, on the lower side of the pencil shaft, is drawn to the back by a spring and strikes a "kicker," which in turn strikes the detent that holds the drum shaft, and causes it to release the shaft. The drums are moved by their own spring, and a mark about three-quarters of an inch long is made on the card. The whole operation is performed in a small fraction of a second. By means of the toothed sector *O*, on the drum shaft, the line is made to occur on the proper section of the card. For pressures below the atmosphere, communication is closed to the main column by the valve *R*, and opened to the vacuum column by the cock *S*. The switch *T*, is shifted to bring the button *P'* into circuit. The connection to the air pump is through the valve and pipe *U*, and a chamber that has about twelve times the cubic capacity of the manifold. A vacuum in the manifold is destroyed by admitting steam through the valve *V*. The scale on the vacuum column is graduated in both pounds and inches, and has at the bottom an arrangement of sliding wedges operated by the lever *W*, raising and lowering the pot, so as to keep the level of the mercury always at the zero of the scale. The small pot *X* in the pipe between the manifold and the vacuum column is a separator to prevent water getting into the column. The vacuum is obtained by a Packard vacuum pump, which will produce a vacuum of within three-tenths of an inch of the barometer reading.

A test proceeds as follows: The indicator is carefully examined to see that the bearing surfaces of the cylinder and piston are clean, and the motion adjusted and lubricated.

Then, with the indicator in a vertical position, and without the spring, the motion is raised and allowed to fall by its own, and the weight of the piston, to see that there is no excessive friction in the motion or between the piston and the cylinder. Then, with the pencil so adjusted that a ray of light can come between its point and drum, the piston is moved back and forth by blowing into, and exhausting from the indicator cylinder, the pencil being observed to see that it moves in a direction parallel to the surface of the drum. With the pencil in the same position the drum is pulled around to see that it is properly centered, and cylindrical. The indicator is then put on the manifold and "harnessed." Steam has in the mean time been blown through the manifold to warm it up. When the steam in the manifold comes through dry the indicator is warmed up and a card put on the drum. Silicated or metallic cards are used, as they permit the use of a metallic point with which a very fine distinct line can be obtained.

After "setting" the machine, the manifold is exhausted to about two pounds below the atmosphere for the purpose of pulling the piston down preparatory to beginning the *up* stroke test. The manifold is then opened to the atmosphere and the *up* zero line is made. The manifold is closed again and steam admitted in sufficient quantity to cause the mercury in the column to rise at the rate of about 2.5 feet per minute. At five or ten pounds intervals, depending upon the scale of the spring, lines are made. When the limit of the test is reached, the pressure is carried one interval higher, and after shifting the drums to bring the down lines in a new position, is allowed to fall, and the down lines are made, the manifold being opened to make the *down* zero line. In testing springs for pressure below the atmosphere, the test is begun at the lower limit, and lines are made at $2\frac{1}{2}$ pounds intervals.

When three tests have been made, a vertical line intersecting each of the horizontal lines is traced with the indicator. The piston is then taken out and calipered to one-ten-thousandth of an inch while hot. The card is taken off and marked with the number and make of the indicator, the number and scale of the spring, the temperature of the mercury column, and the date of the test. Fig. 138 is a fac-simile of a test card.

The cards are measured under a powerful glass with a scale graduated to one-hundredths of an inch; and in measuring, the fractional part of a graduation is estimated in tenths. This brings the readings approximately to one-thousandths of an inch; and in this connection it may be well to state that it is useless to carry the calculation for power further than two places of decimals, as that is beyond the limit of accuracy in measurement. The point from which the measurements are taken in each test, and which is assumed as a zero of the instrument is on the vertical line midway between the up and down zero lines.

The cards are measured by two observers, and, where there is a difference in their readings that cannot be reconciled, the mean is taken.

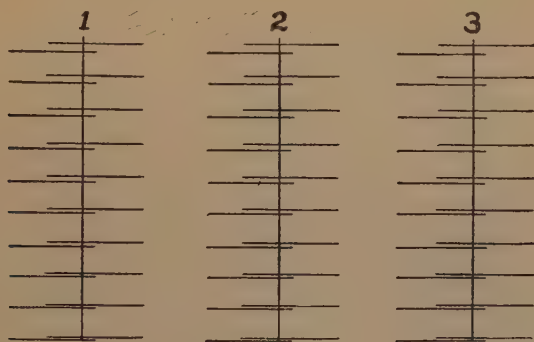


FIG. 138.

The readings are corrected for depression in the level of the mercury reservoir as the mercury enters the column, and also for the dilatation of the mercury due to any departure of the temperature from that at which the scale is graduated.

Correcting Indicator Cards.—The tests of indicators furnish a set of tables for each spring, giving the correct readings both up and down for the different readings taken from the card.

To correct these cards: The card, Fig. 139, *ABCD*, is of the general type of the H. P. and I. P. cards. Draw *AF* and *HE* tangent to the card and perpendicular to the atmospheric line. Then *ABC* is treated as a *down* line, and *CDA* as an *up* line. Now obtain the mean ordinate of the area *ABEFA*. Taking the

area by the planimeter and dividing by the length, or direct by means of the Coffin instrument, this gives the mean uncorrected forward pressure, say, 125 pounds. Look in the *down* column of the table and there find the correct pressure corresponding to 125 pounds of the spring, and suppose that it is 120 pounds.

Proceed in the same manner with the back pressure area *CEFD*, and suppose that its mean ordinate is 62 pounds. Then in the *up* column of the table find the true pressure corresponding to 62 and suppose it to be 60 pounds. Then the corrected mean effective pressure will be $120 - 60 = 60$ pounds.

In a case where special accuracy is required, or where *CK* on the card is decidedly down, the back pressure area should be divided into two or more parts, separating the down from the up lines. Usually, this will not be required.



FIG. 139.

For the L. P. card a different method is necessary, due to the fact that it cuts the atmospheric line and in constructing the tables we have to use the atmospheric line as a zero, from which to start and work both ways.

Draw the tangents as before, Fig. 140. Then obtain the mean ordinate of the area, *ABCDHA*, and apply the *down* correction. Then get the mean ordinate of *CEFGHDC* and apply the *up* correction. The sum of these two will be the corrected mean effective pressure. In measuring the first area we have subtracted *BDC* and added *GFH*, and in the second we have subtracted *GFH*, and added *BDC*, so that these additions and subtractions balance each other. If *BDC* and *GFH* are not exactly equal, a slight error may arise due to the fact that the correction is applied to a pressure that is either slightly greater or less than it should be. But the

error due to this is so small that it can be neglected without materially affecting the results.

The Parallelism of the Pencil Movement.—This is tested by removing the spring and drawing the atmospheric line. Then, on replacing the spring, force the piston upward for several positions of the drum. The lines thus drawn should be parallel to each other and perpendicular to the atmospheric line.

Accuracy of the Drum Motion.—The sources of error in the drum are, from the spring, the inertia, and the friction of the drum. Any increase of tension on the cord from the drum spring lengthens the cord and consequently shortens the diagram. The effect of the inertia of the drum is to lengthen the diagram.



FIG. 140.

Back Lash and Friction.—Draw the piston up, release it, and draw the atmospheric line, then draw another atmospheric line after the piston has been pushed down and released. The difference between the two atmospheric lines will indicate the errors in this direction.

The tendency of abnormal pencil friction on the paper is to cause it to make the pencil lag behind, steam line too low and back pressure line too high, the expansion line insufficiently steep and the compression line too steep.

Indicators cannot, of course, measure the horsepower of a turbine. There have been many different instruments designed to measure the power of turbines, one only of which, a torsionmeter, will be mentioned here. Such an instrument will not give accurate results if applied to a reciprocating-engine shaft, because the twisting moment is not uniform, as it is in a turbine shaft. The Gary-

Cummings Torsionmeter, for measuring the shaft horsepower (*i. e.*, the power actually transmitted to the shaft), will be found on nearly all our recently built turbine vessels; the following excellent description of it is by Commander U. T. Holmes, U. S. N.:

The Gary-Cummings Torsionmeter.—Fig. 141 shows this instrument as supplied to the U. S. S. Florida. A tube *A*, about fourteen feet in length, is installed inside a section of the propeller shafting. One end of this tube has clamped on it a spider casting *B*, as shown at the left and in section at *X-X*. Each leg of this spider carries a spindle *C*, which is a close fit in *B*, with one end turned to a sharp point and the other cut to form a feather that fits in a tapered keyway on screw *D*. The three keyways in *D* are about $\frac{3}{16}$ inch deep at the left end of screw and taper to about $\frac{1}{82}$ inch at the right. A long-handled socket wrench fits nut *E* and is actuated from the left end of shaft. The screw plug *D* is thus drawn to the left, and, since the keyways in which spindles *C* fit are tapered, the points on *C* bite into the shaft, thus binding the spider and the left end of tube rigidly to the inner wall. *F* is a lock washer which serves to prevent *E* working loose.

The right end of tube *A* has a casting *G* clamped on it, and this rests in a ball bearing fitted into a spider set into the end of shaft and held firmly by means of tapered screws *H*.

The arrangement described above permits the tube *A* to remain free from torsion while the shaft is transmitting power. The left end is clamped to shaft, but the right end is permitted to turn freely in its ball bearing and will do so in case there should be an angular displacement of the shaft at this end. It is evident that there will be such a movement and the amount will be proportional to the twisting moment, or torque, transmitted by the shaft.

It remains, then, to provide a means for measuring and recording this movement. A lever *L* is secured to the right end of tube *A* and lies in the slot *S*, cut for this purpose in the face of flange on the shaft. The width of this slot is such as to freely permit the small movement of lever caused by the slight angular movement of tube relative to shaft. A top piece *M* secured to *L* by screw *J* simply serves to transmit the movement of *L* to the recording mechanism. A double arm *T*, made of tubing to give the maximum

strength on minimum weight, is fitted in a block U that is centered at KK and actuated by M . At each end of I is a german-silver recording needle, secured by binding screw not shown in figure. The ball O , section $Y-Y$, is held against screw P by spring Q and pin R , thus providing for taking up all lost motion due to wear.

Since I and O are both firmly secured to the block U moving about $K-K$ as a center, any movement of L with respect to the flange on shaft causes one of the two points NN to go to the left and the other to the right. It still remains to provide means to record this movement, that is, the distance between the circular paths traced by the two points NN as they revolve with the shaft.

At the lower right-hand corner of the figure is shown the device for holding the small sheets of metal-coated paper for taking the record from the needles NN as they revolve in the paths $V-V$. Several sheets of this paper are placed loosely in the pocket provided, and by pushing the rod W the top sheet is moved until it becomes tangent to $V-V$ and receives that record. Z is a handle for moving the paper holder so that if it should be found that only one needle touches, the plane of the paper can be slightly changed so that both needles will mark. S' is an adjusting screw for moving paper holder in and out from center of shaft.

The linear distance between the lines traced by NN is a direct measure of the twist of shaft when corrected for the zero reading. This is obtained by dragging the propeller in order to cause the shaft to revolve when transmitting no power. Both points can be made to mark the same line by careful adjustment of the screw P , but this is inadvisable since the shaft must be stopped to make the adjustment and a zero reading of as much as 0.1 inch is of no disadvantage.

The proportions of the Florida's instruments are such that one-half degree angular displacement of shaft is equivalent to 1.9 inches of displacement between the two recording points. This ratio was found in shop tests with a calibrating instrument especially devised for the purpose. The results of these tests are plotted in the upper right-hand corner of the figure, and show that the ratio between angular displacement and distance between lines on the record is constant for all powers. The tests were made first by

adjusting the arm in the slot, using a micrometer gage to place it in the exact position required. The arm was then locked in this position and the calibrating instrument run in a lathe and cards taken while running.

It was also shown by these experiments that the speed of rotation does not affect the reading of the instrument. Centrifugal force only tends to increase the diameters of the circles traced by the needles, and there is no tendency to vary the distance between the planes of these circles. The linear distance between marks on a card when laid flat is directly proportional to the angular twist of shaft. It is thus seen that one constant serves for all powers and speeds.

To obtain the horsepower constant the shafts are calibrated in the shop by applying a known moment and measuring the angle of twist in a known length by means of battens. Thus in determining the horsepower constant for No. 2 shaft on the Florida the following method was employed:

From calibration data it was found that one-half degree of twist in 100 inches of length of shaft corresponds to 171,371 foot pounds of moment. One inch reading on torsionmeter card corresponds to 0.263 degree of twist. Hence 0.263 degree of twist when the shaft is transmitting power must correspond to $0.263 \times 171,371 \times 2 = 90,312$ foot pounds of twisting moment or torque.

$$\text{S.H.P.} = \frac{2\pi \times M \times \text{R.P.M.}}{33,000},$$

where M = twisting moment in foot pounds. If the torsionmeter reading is one inch we could substitute for M the number 90,312, but the reading varies directly with the torque in all cases, and

$$\text{S.H.P.} = \frac{2\pi \times (90,312 \times T) \times \text{R.P.M.}}{33,000} = 17.195 \times T \times \text{R.P.M.},$$

where T = the corrected torsionmeter reading in inches.

The above expression applies to a length of 100 inches in No. 2 shaft of the Florida. In any shaft the constant is inversely proportional to the length of shaft under consideration. It is therefore possible to cut the tube A of such length that the constant will have a value of 10, and the work of calculating horsepower from the torsionmeter reading can be thus greatly facilitated.

This instrument possesses the advantage of being housed in the shaft, where it is entirely out of the way. There are few parts to get out of order, there is no chance for lost motion to develop and the weight is less than in other torsionmeters. It is applicable only to hollow shafting. The tube and attachments must be placed there permanently and cannot be adjusted or even examined without removing the section of shaft in which installed. The parts are rigidly constructed with locking devices on all pieces that could otherwise work loose. When carefully installed it is believed that there will be no necessity for further adjustment of these parts.

FORMULAE.

The moment M , caused by the pull W on an arm of length $r = W \times r$. From the results of calibration experiments a curve is constructed, the ordinates giving the values of M in *foot pounds* and the abscissae the corresponding readings of the torsionmeter. From this curve the moment corresponding to any reading of the torsionmeter, when the shaft is transmitting power, may be obtained.

The shaft horsepower (S.H.P.) is equal to

$$(1) \quad \frac{2 \times \pi \times M \times \text{R.P.M.}}{33,000} = \frac{M \times \text{R. P. M.}}{5,252}.$$

From applied mechanics we know that the angle of torsion, for a shaft within the elastic limit,

$$(2) \quad \Phi = \frac{2 \times m \times l}{E \times I \times d} = \frac{\Delta}{r} = \frac{\pi \theta}{180}, \text{ where}$$

Φ = the angle of torsion in circular measure;

θ = the angle of torsion in degrees;

Δ = the linear deflection measured on a circumference of radius r ;

m = the twisting moment in *inch pounds*;

l = the length of shaft in inches;

d = the outside diameter of the shaft in inches;

d_1 = the inside diameter of the shaft (if hollow) in inches.

E = the modulus of elasticity of the shaft;

I = the modulus of the section = $\frac{\pi d^3}{16}$ for solid shafts, or

$$\frac{\pi (d^4 - d_1^4)}{16d} \text{ for hollow shafts.}$$

It is more convenient to express the twisting moment in foot pounds, $M = \frac{m}{12}$. Making this change, we have

$$(3) \quad M = \frac{E \times I \times d \times \pi \times \theta}{360 \times 12 \times l} = \frac{\pi^2}{16} \times \frac{E \times d^4}{360 \times 12 \times l} \times \theta.$$

Substituting this value of M in formula (1) for horsepower, we have

$$(4) \quad \text{S.H.P.} = \frac{2 \times \pi \times M \times \text{R.P.M.}}{33,000} = \frac{2 \times \pi^3 \times E \times d^4 \times \theta \times \text{R.P.M.}}{360 \times 16 \times 12 \times 33,000 \times l} \\ = \frac{E}{36,782,000} \times \frac{d^4 \times \theta \times \text{R.P.M.}}{l} \text{ for solid shafts.}$$

Similarly for hollow shafts,

$$(5) \quad \text{S.H.P.} = \frac{E}{36,782,000} \times \frac{(d^4 - d_1^4) \times \theta \times \text{R.P.M.}}{l}.$$

From (3) we have

$$(6) \quad E = \frac{360 \times 12 \times 16 \times l \times M}{\pi^2 \times d^4 \times \theta} = \frac{7,003.2 \times l \times M}{d^4 \times \theta} \\ = \frac{583.6 \times l \times m}{d^4 \times \theta} \text{ for solid shafts;}$$

and similarly,

$$(7) \quad E = \frac{7,003.2 \times l \times M}{(d^4 - d_1^4) \times \theta} = \frac{583.6 \times l \times m}{(d^4 - d_1^4) \times \theta} \text{ for hollow shafts.}$$

From (6) or (7) the modulus may be calculated when the results of calibration of the shafting have been obtained. This has been done for the shafting of a large number of our vessels, and the results are given in the following table:

Ship.	Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Shaft No. 4.	Outside diam., inches.	Inside diam., ins.
<i>North Dakota</i> ...	11,495,800	11,590,800			14	8
<i>Preston</i>	11,825,555	11,926,444	11,825,555		6 $\frac{1}{2}$	3 $\frac{3}{4}$
<i>Flusser</i>	11,548,000	12,100,000	11,730,000		6 $\frac{1}{2}$	3 $\frac{3}{4}$
<i>Reid</i>	11,500,000	12,030,000	11,810,000		6 $\frac{1}{2}$	3 $\frac{3}{4}$
<i>Monaghan</i>	11,550,000	11,675,000	11,360,600		6 $\frac{1}{2}$	3 $\frac{1}{2}$
<i>Terry</i>	11,219,000	11,396,000	11,505,000		6 $\frac{1}{2}$	3 $\frac{1}{2}$
<i>Chester</i>	11,622,702	11,820,000	11,840,000	11,755,405	8 (solid)	
<i>Salem</i>	11,378,947	11,687,837			9 $\frac{3}{4}$	5
<i>Perkins</i>	11,490,000	11,551,000			8	4
<i>Sterett</i>	11,484,000	11,635,000			8	4
<i>McCall</i>	11,685,887	11,891,530	11,825,900		6 $\frac{3}{4}$	3 $\frac{3}{4}$
<i>Burrows</i>	11,952,038	11,895,519	11,878,224		6 $\frac{3}{4}$	3 $\frac{3}{4}$
<i>Florida</i>	11,695,300	11,659,600	11,616,600	11,844,600	12 $\frac{1}{2}$	6
<i>Arkansas</i>	11,936,903	11,962,910	11,893,953	11,818,720	12 $\frac{1}{2}$	8
<i>Teet shaft</i>	11,577,071				1 $\frac{1}{2}$ (solid)	

These results do not show great variation, but the difference is sufficient to justify the rule that all shafting on which a torsionmeter is to be used must be calibrated. It is probable that the results would have approximated more closely to a mean value if the axial hole had in all cases been exactly central and of the correct diameter throughout, and also if the experiments had been carried out under the same conditions by the same observers.

When calibration is impracticable, approximately correct results in the case of high-grade steel shafting will be obtained if we assume a value for E of 11,750,000. Substituting this value in (4) and (5) we have

$$(8) \quad \text{S.H.P.} = \frac{d^4 \times \theta \times \text{R.P.M.}}{3.13 \times l} \text{ for solid shafts, and}$$

$$(9) \quad \text{S.H.P.} = \frac{(d^4 - d_1^4) \times \theta \times \text{R.P.M.}}{3.13 \times l} \text{ for hollow shafts.}$$

CHAPTER XI.

CURVES OF CRANK EFFORT.

While the indicator gives by means of a diagram a measure of the effective pressure exerted by the steam on the piston, the manner in which it is transmitted to the shaft is yet to be determined. As the rate of work on the crank is continually varying during a revolution, and steadiness of driving depends upon the uniformity of the turning effort, it is important to know the nature and extent of this variation in order to provide means of reducing it to a minimum. This may be studied by means of a *diagram of crank effort* showing the work done on the crank, together with the twisting and bending stresses brought upon it.

The *turning* force is that component of the force transmitted through the connecting rod (when resolved tangentially and radially) which tends to rotate the crank. By ascertaining the values of this turning force for different crank positions during a revolution of the engine, laying them off on the corresponding radial extensions on a crank pin circle and connecting the extremities by a curved line, the resulting figure is the *polar* or *circular diagram of crank effort*, from which the ratio of the maximum to the mean twisting moment can be found for use in calculating the required size of shaft.

In laying out this diagram the following considerations producing the variable turning effort must be considered:

1. The pressure of steam acting on the piston varies for different points of the stroke.
2. By the employment of the crank and a connecting rod of finite length, the variable piston pressure is transmitted through a connecting rod which meets the crank at a constantly changing angle. This force may be resolved into two components, one in the direction of the crank and the other, at right angles. The tangential component alone produces any turning effort; the radial component simply exerts pressure on the bearings without

producing any movement. The resulting turning effort is variable, as it depends upon the varying angles of inclination of connecting rod and crank and varies from zero to a maximum, twice during a crank revolution.

3. In addition to the above, the turning effort depends upon the inertia of the reciprocating parts of the engine, piston, piston rod, crosshead, and connecting rod, which absorb power in acquiring motion or velocity during the early part of the stroke, while in the latter part of the stroke it is restored to the crank pin, as these parts gradually come to rest.

1: Taking the simplest case, that in which the piston pressure is uniform throughout the stroke, connecting rod infinite in length, and neglecting the weight of the reciprocating parts:

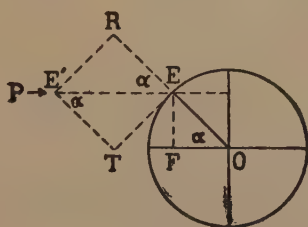


FIG. 142.

In Fig. 142, OE = crank position, P = piston pressure. Resolve P radially and tangentially into RE and TE . Turning force = $TE = EE' \sin \alpha = P \sin \alpha$. The twisting or turning moment on the crank shaft is equal to the tangential pressure multiplied by the leverage at which it acts, namely, the length of the crank; hence, the twisting moment at the point $E = P \times EO \sin \alpha$. Since the length of the crank is constant, the diagram of tangential pressures may be made to represent that of twisting moments but to a different scale, the former representing pounds and the latter, pounds multiplied by the length of the crank in inches.

From the above it can be seen that although the piston pressure may be uniform throughout the stroke, the turning effort is very variable, zero at commencement, a maximum when the crank is at right angles to the axis of the cylinder and zero again at the end.

Although the average tangential pressure is less than the piston pressure, the work done upon the crank is exactly equal to that done upon the piston (neglecting friction), for the path described by the crank pin is greater than that traversed by the piston in a double stroke in the ratio of π to 2.

In one stroke work done by piston = $P \times \text{stroke}$.

In one stroke work done by crank = $P \frac{2}{\pi} \times \frac{\pi \times \text{stroke}}{2}$
 $= P \times \text{stroke}.$

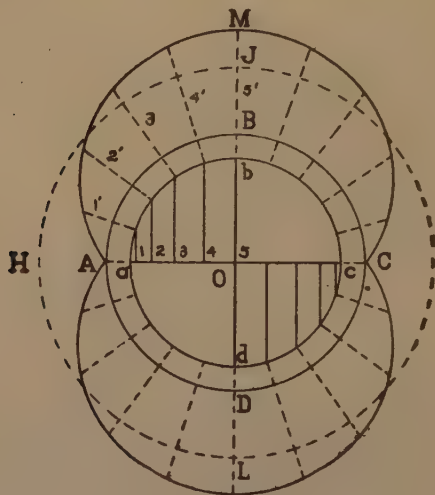


FIG. 143.

Graphic Representation of the Crank Effort.—From the center O , Fig. 143, draw the circle $ABCD$ to represent the crank pin circle and the circle $abcd$ with a radius equal to the uniform piston pressure assumed. Divide the circumference of the pressure circle into any number of equal parts and let fall perpendiculars upon the diameter, from the points of division. Since at each position of the crank represented by the points of division, the turning force is equal to the pressure on the piston multiplied by the sine of the crank angle. Since Oa is drawn to represent the piston pressure, the perpendiculars 1, 2, 3, 4, etc., represent to scale the tangential or turning force for the different crank positions. Transfer the ordinates 1,

2, 3, 4, etc., to the prolongation of the radial lines beyond the crank pin circle, 1', 2', 3', 4', etc., and joining the extremities by the curved line AMC , the tangential pressure or turning effort is represented graphically and accurately at any desired crank position in the forward stroke. For the return stroke, proceed in like manner in the lower half of the crank circle.

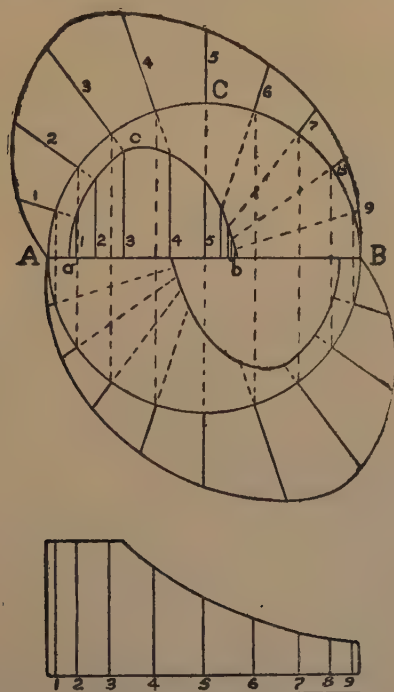


FIG. 144.

The Mean Tangential Pressure may be represented by a circle drawn with a radius OH , and is obtained from $HA : P$ or $Oa : 2 : \pi$, when P is the mean effective pressure on the piston. With an engine running at a uniform rate of speed this circle is known as the *resistance circle*, as the mean tangential pressure is exactly equal to the resistance of the load to the motion of the crank pin.

2. When the Pressure Varies as in an Expansion Engine.—As before, assume connecting rod infinite in length and reciprocating

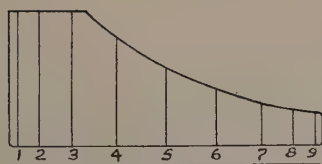
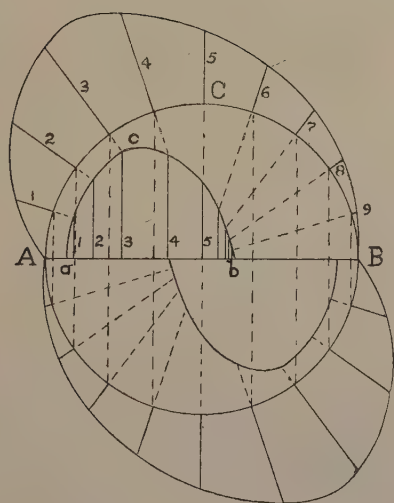


FIG. 144.

It is frequently found more convenient to use the *linear diagram of effort* instead of the polar diagram, and may be illustrated by unrolling the crank circle $ABCD$, Fig. 143, on a straight base, when the diagram will take the shape shown in Fig. 145. To construct this diagram draw a straight line AB equal to the semi-circumference of the crank pin circle and divide this line into the same number of equal parts corresponding to the division of the semi-circumference ACB . At the points of division set off ordinates $1-1'$, $2-2'$, etc., equal in length to the lines 1, 2, 3, etc., in Fig. 133. Through the extremities of the ordinates $1-1'$, $2-2'$, etc., draw the curve $A-1'-2'-B$, and the ordinates of this curve will show the tangential pressures on the crank for the different crank positions during the half revolution.

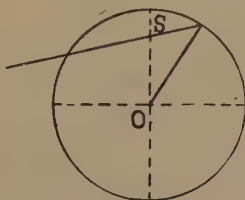


FIG. 146.

Since the ordinates in Fig. 145 represent effective crank pin pressures, and the abscissae, the corresponding distances moved through, the area ACB represents the work done by the crank, and, neglecting friction, should be exactly equal to the area of the indicator diagram from which measurements were taken. For the return stroke prolong AB a distance equal to its length and construct the curve as before.

3. The Influence of the Connecting Rod.—In the study of “Elements of Mechanism” it has been shown that if the connecting rod be drawn to any position of the crank, and if the crank arm represent the magnitude of the steam pressure, then on the same scale, the distance OS , Fig. 146, intercepted between the center and the direction of the rod will give the crank effort for the given position.

As shown in the earlier chapters, the effect of the connecting rod is to increase the movement of the piston during the first

circular diagram, the length of the corresponding ordinates from the indicator card; for instance, when the crank is at position 8, lay off OS equal to the steam pressure at position 8 of the card. When the crank is at this position, OF represents the crank effort on the same scale that OS' represents the magnitude of the steam pressure. Now, if through S , SS' be drawn parallel to $8F$, then $\frac{OS'}{OS} = \frac{OF}{OS}$ and since OS represents the pressure taken from the card, the crank effort on the same scale is represented by OS' . Lay off an equal distance 8-8" radially from the circumference of the crank circle. In the same manner find values of the crank effort for the different crank positions and transfer these values to the radial ordinates 1-1", 2-2", 3-3", etc. Through these extremities draw the curve AMB , for the curve of tangential crank effort.

The Influence of Inertia of the Reciprocating Parts.—In starting the reciprocating parts from a state of rest at the beginning of the stroke to their maximum velocity, a large portion of the energy of the steam on the piston may be absorbed in overcoming the inertia of those parts, leaving only a portion of the steam pressure on the piston available for transmission to the crank pin, while during the latter part of the stroke this power absorbed in acceleration is restored during retardation, the effect of reducing the velocity of the reciprocating parts again to zero, causing a greater pressure on the crank pin than is due to the pressure on the piston. Under these circumstances before constructing the diagram of the crank effort the indicator diagram must be corrected by modifying the effective pressures for the various points of the stroke, due to the inertia of the reciprocating parts.

Neglecting friction, the total work upon the crank pin is not affected, since the power absorbed during acceleration is equal to that restored during retardation, but the distribution of the turning effort is changed and the effect upon the steadiness or regularity of the engine may be very great.

The Force Required to Produce Acceleration.—With a connecting rod infinitely long it has been proved that the velocity of the piston at any point C , Fig 148, is proportional to the sine of

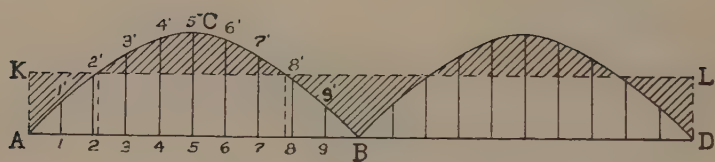


FIG. 145.

The force required to produce this acceleration $= F = \text{mass} \times \text{acceleration} = \frac{Mv^2}{r} \cos \theta$. Since the expression for centrifugal force $= \frac{Mv^2}{r}$, F is the horizontal component of the centrifugal force of a mass M concentrated at the crank pin; and since, in the second quadrant, cosine is negative, F acts as a retarding force.

Substitute for M its value $\frac{W}{g}$ and for v its value $\frac{2\pi r N}{60}$, where $N = \text{revolutions per minute}$. Accelerating force $= \frac{W}{gr} \left(\frac{2\pi r N}{60} \right)^2 \times \cos \theta = .00034 WN^2 r \cos \theta$ and the pressure, per square inch of piston area, to produce this acceleration $= \frac{.00034 WN^2 r \cos \theta}{A}$, where $A = \text{area of piston}$.

Since, in moving from the dead center in the first quadrant, the pressures formed go towards accelerating the moving parts, they must be subtracted from the pressures on the indicator card, for the corresponding points, in order to obtain the proper turning effort. Similarly, in the second quadrant, the reciprocating parts are retarded. Therefore, pressure is given to the crank pin, and the values found must be added to the corresponding indicator pressures. For example, in Fig. 149, let the engine work with a constant pressure of 30 lbs. of steam. Draw AB the same length as the card. Calculate by the formula $.00034 WN^2 r$ the centrifugal force per square inch of piston area, for the known weight of the reciprocating parts; radius of crank circle; and number of revolutions per minute. Assume the calculation to give ten pounds per square inch of the piston, as that required at the commencement of the stroke, to accelerate the reciprocating parts.

From A lay off AC , the steam pressure $= 30$ lbs., $AD = 10$ lbs. Through E , the center of AB , draw the line DEF . Then the ordinates between A and E require to be subtracted from, and those between E and F to be added to, the pressure given by the card, and the diagram now takes the form $DFGC$, showing that the actual pressure imparted to the crank pin is but 20 pounds at the commencement, 40 pounds at the end of the stroke, and at E , the mid-position of the piston, the acceleration is zero.

Influence of Weights of Reciprocating Parts in Vertical Engines

—In the case of vertical engines, Fig. 149, must be modified to take into account the dead weight of the reciprocating parts. During the up stroke these weights act against the piston and their influence is allowed for by drawing an additional horizontal line above AB , as in Fig. 150, at a distance AH' equal to the weight of the reciprocating parts per square inch of piston area. The new base line is now $H'C'$ and is the base of the area $H'D'E'$ whose ordinates must be subtracted from the corresponding ordinates of the pressure diagram, while those of the area $F'C'E'$ must be added in order to obtain the turning effort on the crank pin. For the down stroke the conditions are reversed, the weight of the reciprocating parts acting *with* the steam pressure, the accel-

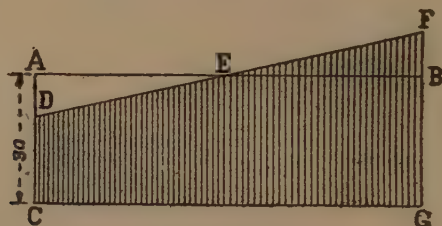


FIG. 149.

erating force required, Fig. 151, at the beginning of the stroke A , would be equal to HD , where AH as before represents the weight of the reciprocating parts in pounds per square inch of piston area, the ordinates of the area HDE are consequently subtracted from, and those of FEC added to the corresponding ordinates of the pressure diagram to obtain the turning effort on the crank pin for the down stroke.

The Influence of the Angularity of the Connecting Rod on the Reciprocating Parts.—When there is a connecting rod of finite length the inertia diagram is no longer bounded by a straight line, DF , Fig. 151, but becomes a curve of somewhat complex character, the ordinates of which represent the pressures of acceleration and retardation for corresponding crank angles.

To Ascertain these Pressures.—When the crank pin is at the dead center, the reciprocating weights act with a full centrifugal force

on the crank. At the near center they pull the crank towards the cylinder, and at the opposite, the reverse. The centrifugal force at the near center gives the measure and direction of the force absorbed in accelerating the reciprocating parts, with an infinite rod. With a finite rod, however, on passing the near center the crank pin end of the connecting rod describes a small portion of the arc of a circle with the piston rod end as a center, and centrifugal force is produced which increases that of the crank. On passing the far dead center this centrifugal tendency of the connecting rod end diminishes that of the crank. A rough approximation of the amount of the connecting rod centrifugal force, has been given by a number of writers, and has been derived from that of the crank, as the weights and velocities are the same and only

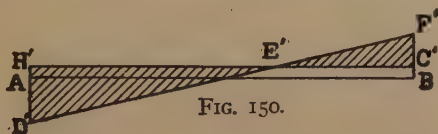


FIG. 150.

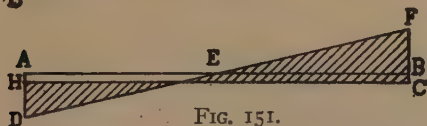


FIG. 151.

the radii differ. Other things equal, the centrifugal force varies inversely as the radii, hence, the centrifugal force of the connecting rod at the dead centers is that of the infinite rod divided by the ratio of the connecting rod to crank. So that at the near dead center the ordinate equals centrifugal force multiplied by $(1 + \frac{\text{crank}}{\text{rod}})$, and at far end the ordinate equals centrifugal force multiplied by $(1 - \frac{\text{crank}}{\text{rod}})$. For a third point of the curve, works on mechanics give methods of plotting curves of piston velocity, from which can be found the maximum velocity of the piston and the corresponding ordinate, at which point the acceleration is zero.

For a connecting rod greater than three times the crank, the position of maximum velocity is very nearly that position where the connecting rod is tangent to the crank pin circle. For this position

$\frac{\text{length of connecting rod}}{\text{length of crank}} = \text{tangent of crank angle}$. By drawing a fair curve through the three points thus found the inertia line may be obtained sufficiently accurate for all practical purposes. As before, the ordinates of the curve between *C* and *H*, Fig. 152, represent pressures expended in accelerating the reciprocating parts, and must be subtracted from the card; and between *D* and *H*, ordinates represent pressures brought on the crank by the moving parts, and must be added to the indicator card. The length of the intermediate ordinates of this curve can be obtained, if desired, from the equation of the inertia curve, page 283.

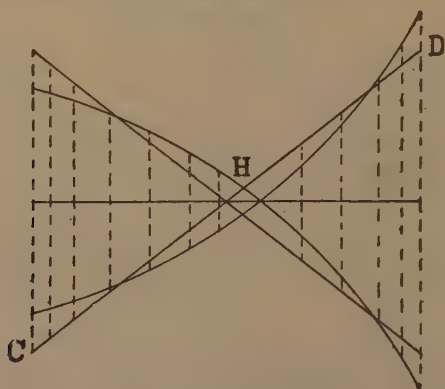


FIG. 152.

Correction for Indicator Cards.—To obtain the true effective pressure, diagrams from both ends are necessary, as for each ordinate representing the forward pressure, the corresponding back pressure on the opposite end must be deducted to obtain the true effective pressure. So for a proper computation of crank effort, the two diagrams must be combined and a new diagram formed. This true diagram of resultant pressures is formed by laying off as ordinates the difference between the forward pressures at one end and the back pressures on the other end, measuring from the vacuum line, and drawing a curve through the ends of the ordinates. When the forward pressure is in excess, the ordinate is above the vacuum line, and when it is less than the back pressure it is below the vacuum line. In Fig. 153 cards are shown from the

I. P. cylinder of the Brooklyn, in which the true diagram of effective pressures for the top end is represented by the shaded area, while that for the opposite end is shown by the dotted lines. AB is the atmospheric line. At G the forward pressure is GC and at the same time there is a counter pressure measured by GH , so that the effective pressure equals the difference GI . By applying this operation to each ordinate the corrected diagram $BIEFAB$ is obtained. At E , the forward pressure equals the back pressure, the piston is carried on by the momentum of the moving

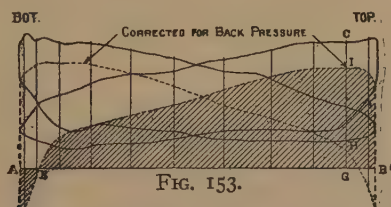


FIG. 153.

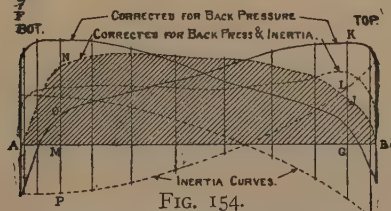


FIG. 154.

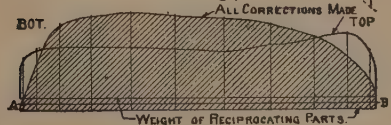


FIG. 155.

parts, and from this point to the end of the stroke the forward pressure is less than the back pressure, the curve becomes negative and the diagram passes below the base line as shown by the area AEF .

In Plate 27 is given the diagram for each crank of the Brooklyn, laid out in accordance with the principles just explained. Figures 153, 154 and 155 show in detail the corrections applied for back pressure, for inertia, and for weight of the reciprocating parts from the data given in Plate 27, Fig. 155, giving the final

diagram, from which are taken the distances to be laid off on the corresponding radial ordinates shown in the plate.

In Fig. 154 each ordinate is corrected for inertia by subtracting from the corrected curve in Fig. 153, the distance between the inertia curve and base line, for the early part of the stroke, and adding this difference for the latter part of the stroke. For instance, to obtain the point *J* on the curve corrected for inertia, $GJ = GK - GL$, and for the point *N*, $MN = MO + MP$.*

In this discussion of the subject the inertia taken into account is the resistance to acceleration in the direction of the piston's motion; viz., the translational resistance of the piston and connecting rod. In addition to this there is the resistance due to angular acceleration on the part of the rod, and also the centrifugal force of the connecting rod considered as rotating about the cross-head.

From an examination of the preceding diagrams it can be readily seen how variable is the turning effort upon the crank pin with a single cylinder engine and how the steadiness of the driving power of the engine must be affected. To overcome this irregularity of driving, a fly wheel may be employed, an adjustment of cut-off and compression to suit the speed of the engine, or by the use of two or more engines on the same shaft with their cranks at angles with each other, and set so that the tangential effort on one crank is at a maximum while that on the other is at a minimum, and thus a more uniform turning effort obtained.

Where two or more cylinders are employed on the same shaft, crank effort diagrams are made from each engine in the manner already described and set at the same angles that the cranks make with each other. The sum of tangential pressures at any point in the revolution is found by adding the lengths of the corresponding ordinates for each cylinder and the curves connecting the

*In all the discussion of the indicator cards and the diagram constructed from them we have started by dividing the crank circle into equal parts, and determining the corresponding positions of the piston. It should be understood that the work can be done as well by dividing the length of the indicator card, representing piston positions, into equal parts and determining the corresponding crank positions. It is usual in such cases, to divide one or two end spaces into half spaces to avoid having large angular distances, near the dead centers on the crank effort diagram, without ordinates.

ends of such combined ordinates give the curve of tangential effort of the combined cylinders, as shown in Plate 28. In the case of compound engines the figure must be expressed in similar terms of pressure by adopting a scale for the indicator cards in proportion to the capacity of the cylinders. For instance, if there are two cylinders in the ratio of 1 to 4, each pound of pressure per square inch on the larger piston acts with four times the pressure on the crank pin as one pound per square inch on the smaller piston.

To Show the Effect of Several Cranks.—Using for convenience a straight base as described on page 271, Fig. 145 shows the crank

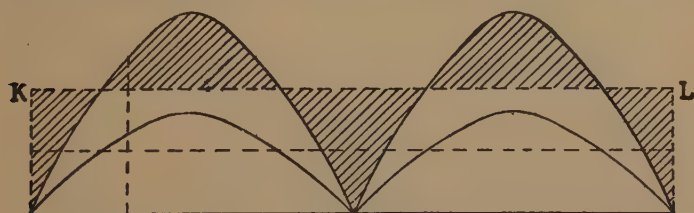


FIG. 156.

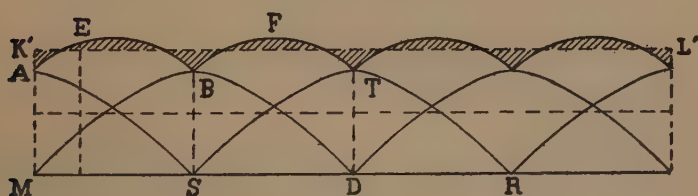


FIG. 157.

effort for a single engine on a single crank. With two engines on the same shaft and with cranks opposite, since each engine is on the dead center at the same time, the maximum and minimum efforts coincide, and are added together, producing a diagram shown in Fig. 156. With two cranks at right angles the range effort is much reduced as shown in Fig. 157, the curve *MBD* being constructed for one engine, and for the next engine the curve is commenced at *S* half way between *M* and *D* or 90° ahead of *M*; *MBD* and *STR* are the diagrams of the separate engines and *AEBFT* is the combined diagram while the mean effort is represented by *K'L'*.

With triple engines and cranks 120° apart the curve commences as before at *M*, Fig. 158, for the first engine, at 120° for the second and 240° for the third. The combined turning effort at any point *A* is given by the sum of the turning moments at that point, $AB + AC + AD$ and the curve drawn as shown, *KL* being the mean turning effort. From the above diagram, the great reduction in the range of the turning effort by the employment of several cranks may be seen, and consequently the greater uniformity in the driving power of the engine.

Twisting Moments.—We have been considering in the foregoing, the turning force or tangential force. The twisting moment is simply the tangential force multiplied by its arm and the curve of twisting moment can be laid off as described for turning force



FIG. 158.

It is customary to represent the twisting moment in foot-tons, and it can be obtained for each angular position by multiplying the turning force by its arm and dividing by 2,240, or, more generally:

$$\text{Twisting moment} = \frac{P \times \text{area of piston} \times C \times \sin(\theta + \phi)}{\cos \phi \times 2,240}$$

P = pressure per square inch (corrected). *C* = length of crank in feet. θ = crank angle. ϕ = angle made by connecting rod with line of centers, Fig. 159. When two or more engines are connected to the same shaft, the combined curve of twisting moment is drawn as explained previously. The twisting moment equals the thrust on the connecting rod multiplied by the perpendicular distance from the center of the shaft to the line of the direction of the connecting rod for the given crank position. Thrust on

connecting rod $= T = \frac{P}{\cos \phi}$. Twisting moment $= \frac{P}{\cos \phi} \times OB$ and

$OB = r \sin(\phi + \theta)$. $\therefore$ Twisting moment $= \frac{P \times \text{area} \times r \sin(\theta + \phi)}{\cos \phi \times 2,240}$

in tons. Since the sines of angles are proportional to the opposite sides, having given the length of the crank and of the connecting rod, and the crank angle θ , the angle ϕ can be easily determined.

For instance, let ratio $\frac{\text{crank}}{\text{rod}} = \frac{1}{4}$. To find ϕ when $\theta = 45^\circ$. $\frac{\sin \phi}{\sin \theta} = \frac{1}{4}$.

$\therefore \phi = \sin^{-1} \left(\frac{\sin \theta}{4} \right) = 10^\circ 11'$.

The middle point of the inertia curve with a finite connecting rod is at a distance from the center of the stroke equal to

$$\sqrt{\text{rod}^2 + \text{crank}^2} - \text{rod}.$$

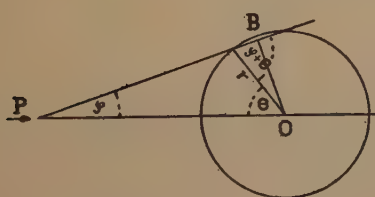


FIG. 159.

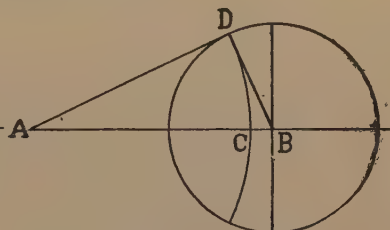


FIG. 160.

The acceleration is zero when the crank and connecting rod are at right angles. With a radius equal to the length of the connecting rod, Fig. 160, describe DC , and C is the position required. $AB = \sqrt{AD^2 + DB^2}$, $CB = AB - AC$ or AD . $\therefore CB = \sqrt{\text{rod}^2 + \text{crank}^2} - \text{rod}$.*

* **The Equation of the Inertia Curve.**—Let ω = uniform angular velocity of the crank, t = time taken by crank to pass from A to P (Fig. 161), x = part of stroke performed in time t , n = ratio of connecting rod to crank $= \frac{1}{r}$.

From the figure, $TQ = TA + AO - BO - QB$ or:

$$\begin{aligned} x &= nr + r - r \cos \theta - \sqrt{n^2 r^2 - r^2 \sin^2 \theta} \\ &= r(n + 1 - \cos \omega t - \sqrt{n^2 - \sin^2 \omega t}) \end{aligned} \quad (1)$$

$$\frac{dx}{dt} = \omega r \left(\sin \omega t + \frac{\sin \omega t \cos \omega t}{\sqrt{n^2 - \sin^2 \omega t}} \right) \quad (2)$$

$$\frac{d^2x}{dt^2} = \omega^2 r \left(\cos \omega t + \frac{\cos^2 \omega t - \sin^2 \omega t}{\sqrt{n^2 - \sin^2 \omega t}} + \frac{\sin^2 \omega t \cos^2 \omega t}{(n^2 - \sin^2 \omega t)^{\frac{3}{2}}} \right) \quad (3)$$

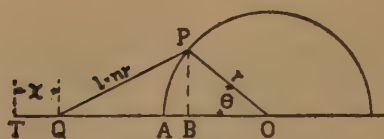


FIG. 161.

When $\omega t = 0$, $\sin \omega t = 0$, and $\cos \omega t = 1$. Therefore $\frac{d^2x}{dt^2} = \omega^2 r \left(1 + \frac{1}{n}\right)$.

When $\omega t = 180^\circ$, $\frac{d^2x}{dt^2} = \omega^2 r \left(1 - \frac{1}{n}\right)$.

These are the accelerations at the ends of the stroke. By giving ωt values from 0° to 180° , the accelerations can be calculated for intermediate points.

The inertia forces corresponding to these accelerations require a factor for mass. The inertia forces per square-inch of piston area are wanted, rather than the whole force; and they are calculated, then, from w , the weight of the reciprocating parts per square inch of piston area and not from W , the whole weight.

Let N = revolutions per second. Then $\omega = 2\pi N$ and the factor $\frac{(2\pi)^2}{g} = 1.24$.

The equation of the inertia curve is, therefore:—

$$F = 1.24 w N^2 r \left\{ \cos \omega t + \frac{\cos^2 \omega t - \sin^2 \omega t}{\sqrt{n^2 - \sin^2 \omega t}} + \frac{\sin^2 \omega t \cos^2 \omega t}{(n^2 - \sin^2 \omega t)^{\frac{3}{2}}} \right\} \quad (4)$$

and the values at the ends of the stroke are:—

$$1.24 w N^2 r \left(1 + \frac{1}{n}\right), \text{ and } 1.24 w N^2 r \left(1 - \frac{1}{n}\right).$$

CHAPTER XII.

DRAINAGE—PUMPING AND FIRE SYSTEM.

The *drainage system* consists of piping with tanks or reservoirs for collecting the water which has entered the ship from leakage, accident, etc., into convenient places for pumping overboard.

To free the ship of water due to the above leakages or damage to the hull done in action, or from accident, and for the distribution of both fresh and salt water through the ship, for flushing, sanitary and fire purposes, various steam pumps are used, which together with a few auxiliary hand pumps constitute the *pumping system*.

Although the pumping system is made as complete and extensive as possible, a simple calculation will show the inadequacy of the pumps alone to rid the ship of water as fast as it enters through a large hole below the water line. The amount of water entering a ship through an opening A square feet in area, and h feet below the water line is obtained as follows:

Velocity of flow into the ship $= \sqrt{2gh} = 8\sqrt{h}$ feet per second.

Quantity of water flowing into a ship $= 8A\sqrt{h}$ cubic feet per second.

Taking the maximum pumping capacity of the battleship at about 5400 tons per hour, or about $1\frac{1}{2}$ tons per second, the largest leak that could be kept under, say 16 feet below the water line, by the pumping capacity of the ship would be,

$$8A\sqrt{16} = 1.5 \times 35$$

$$A = 1.64 \text{ sq. ft.}$$

The necessity for the water-tight subdivisions of the ship is here demonstrated, together with the care that should be taken in time of accident to cut off the damaged compartment from all others. As the injured compartment fills, the velocity of flow decreases, when temporary stoppage can possibly be effected from the outside. In this case, the compartments can be pumped out and temporary repairs made from the inside, sufficient to answer the purpose until arrival in port.

To enable the pumps to reach the various compartments of the ship there must be a system of suction pipes connected with manifolds leading to the pumps by which the water can be removed.

A *manifold* is an arrangement whereby a pump is made capable of drawing from several compartments, and consists of a rectangular valve chest or suction box *A*, Fig. 162, connected to the pump by means of a suction pipe *B*. Pipes *C C C*, from the various compartments lead into the bottom of this valve chest, and their upper ends are fitted with screw-down valves which can be operated at

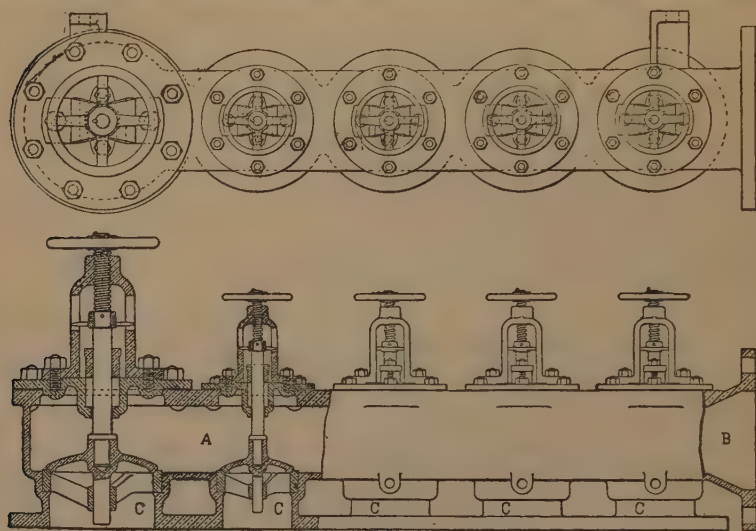


FIG. 162.

the valve chest and sometimes from the deck above. To put any compartment into communication with the pump, all that is necessary is to open the valve which is over the end of the pipe from that compartment, the others remaining closed. Discharge manifolds are arranged in a similar manner.

Three different kinds of valves will be found used in these manifolds, viz.: (1) *lift-check-stop* valves, which have the stem so fitted to the valve disc that when partially unscrewed the valve acts as a *check valve*, i. e., it opens under pressure from below and closes under pressure from above; when fully opened the valve disc is *lifted*

off the seat and water is free to pass in either direction; (2) *lift-stop* valves, which are so fitted that the least movement of stem in the direction of opening lifts the disc off its seat; and (3) *check-stop* valves, fitted so they act only as check valves, and the discs cannot be *lifted* off their seats. All three kinds are held closed by screwing down on their stems.

It is necessary to fit strainers to the bilge suction pipes of all pumps in order to prevent the entrance of anything which would choke the pipes and valves of the pump. The *Macomb Strainer* shown in Fig. 163 has been used for many years in the ships of our navy and consists of the following parts:

The pump suction is attached to the flange *L*. The pipe *D* leading to the bilge is connected to the upper part of the casing, the bilge water passing into the strainer basket *K*, which arrests the contained debris, permitting the water to pass into the lower part of the casing and to the pump suction *L*.

The basket is made of light copper, perforated, and rests on an interior flange in the casing. At intervals the basket is removed and cleaned by slackening the screws *A*, throwing the dogs *B* to one side, removing the cover *G*, and lifting out the basket *K*, by the handle *H*. A spare basket is then inserted and the parts connected up again, the basket taken out being cleaned and prepared for use to replace the spare basket when choked.

In case of serious damage to the hull, the circulating pumps of the main engines are depended upon as the principal factor in keeping the influx of water under control. In the navy these pumps are independent and worked by separate engines, so that the speed is entirely independent of that of the main engines, and in addition

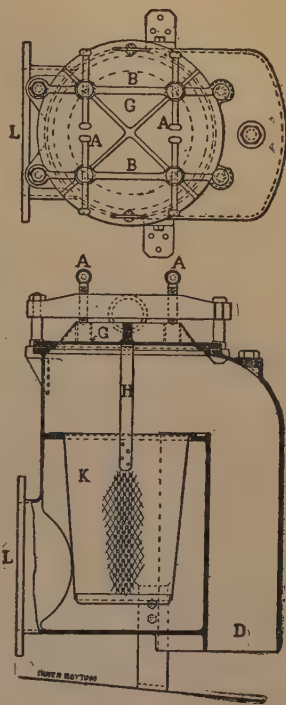


FIG. 163.

TABLE VI.
TYPICAL INSTALLATION OF ENGINE DEPARTMENT PUMPS ON A BATTLESHIP.

No.	Pumps.	Size (inches).	Kind.	Suction pipes from	Discharge pipes to	Location.
4	Main air.	12 by 25½ by 21.	Twin vertical.	Ins. 10 Condenser.	Ins. 8½ Feed tank.	Two in each pump room.
4	Main circulating.		Vertical, compound, centrifugal.	21½ 15½ Sea. Drainage.	21½ 5 Condenser. Oil cooler and water service.	Two in each pump room.
4	Main feed.	15½ by 9½ by 24.	Vertical, piston, double-acting, single.	5½ Feed-tank standpipe cross connection.	5 Main feed discharge.	Two in each pump room.
3	Auxiliary feed.	15½ by 9½ by 24.	Vertical, piston, double-acting, single.	5½ 4 2½ Feed-tank cross connection. C. and R. pipes from ship's side. ¹ Reserve feed tanks. Hose connection.	5 2½ Main feed main. Hose connection. Reserve feed tanks. ¹ Each boiler in the compartment in which pump is located.	One in each boiler room.
4	Main fire and bilge.	12 by 10 by 18.	Vertical, piston, double-acting, single.	5 5 1½ 4 Sea. C. and R. drainage. Hose connection. ² Hose connection. ²	5 5 5 2½ 4 Fire main. Overboard. Sanitary. ² Hose connection. Hose connection. ²	Two in each pump room.
3	Fire-room fire and bilge.	12 by 10 by 18.	Vertical, piston, double-acting, single.	5 5 1½ 4 Sea. C. and R. drainage. Hose connection. ³ Hose connection. ³	5 5 5 2½ 4 Fire main. Overboard. Sanitary. Hose connection. Hose connection. ³	One in each boiler room.
1	Dynamo condenser circulating.		Centrifugal.	Sea.	Dynamo condenser.	One in dynamo condenser room.
1	Dynamo condenser air.		Vertical, twin-cylinder, beam.	Dynamo condenser.	Dynamo-room hot well.	One in dynamo condenser room.

¹ No. 2 boiler room only.

² One pump in each pump room.

³ No. 1 boiler room only.

TABLE VI.—Continued.
TYPICAL INSTALLATION OF ENGINE DEPARTMENT PUMPS ON A BATTLESHIP.

No.	Pumps.	Size (inches).	Kind.	Suction pipes from		Discharge pipes to		Location.
				<i>Inch.</i>		<i>Inch.</i>		
1	Dynamo-room hot well.	4½ by 5 by 6.	Vertical, piston, double-acting, single.	2½	Dynamo-room hot well tank.	2	Auxiliary feed-suction main.	One in dynamo condenser room.
2	Evaporator feed.	5½ by 6 by 6.	Vertical, piston, light service, double-acting, single.	2½	Sea. Overboard discharge from distillers.	2	Evaporators, through feed heaters.	Evaporator room.
2	Distiller circulating.	12 by 14 by 16.	Vertical, piston, double-acting, single; or turbine.	6½	Sea.	6 4½	Distillers. Fire main.	Evaporator room.
2	Distiller fresh water.	4½ by 5 by 6.	Vertical, piston, light service, double-acting, single.	2½	Bottom of distillers. Test tank.		Fresh-water tanks. Main feed tanks. Reserve feed-water tanks.	Evaporator room.
2	Forced lubrication service and return.	6 by 7 by 8.	Vertical, piston, double-acting, single.	3½	Supply tanks.	3	Main engines.	One in each engine room.
2	Forced lubrication service.	6 by 7 by 8.	Vertical, piston, double-acting, single.	3½ 3½ 2	Supply tanks. Oil well. Bottom of oil well.	3	Settling tank and reserve tank. Overboard.	One in each engine room.
2	Forced lubrication return.	7½ by 8½ by 8.	Vertical, piston, double-acting, single.	4 3	Oil well. Reserve tank.	3½	Settling tank and reserve tank.	One in each engine room.
6	Fuel-oil service.	5½ by 4 by 8.	Vertical, piston, double-acting, duplex.	3	Fuel-oil main.	2½	Burner supply.	Two in each fire-room.
4	Fuel-oil booster.	6 by 8 by 8.	Vertical, piston, double-acting, single.	5	Tank manifold.	5 5	Fuel-oil main. Tank manifold. Overboard.	Two in each end fireroom.

to the suction from the sea, there is a bilge suction as shown in Fig. 93, so that the entire suction may be taken from the bilge. A pass valve is also fitted in the water chamber at one end of the condenser so that the water if desired may be discharged directly overboard instead of through the condenser tubes, as shown in Plates 24 and 25. As the lift required in freeing a ship of water, or in passing it from the sea through the condenser is very small, circulating pumps are of the centrifugal type, because of the great quantity of water that can be delivered from pumps of this pattern. In the battleships Louisiana and Connecticut there is one centrifugal pump for each condenser and each of sufficient power to discharge 12,000 gallons of water per minute from the bilge, the revolutions not to exceed 245 per minute. With these two pumps in operation it is possible to discharge 5500 tons per hour.

As the fans have no suction power these pumps will not draw unless charged with water at starting, and kept supplied while in operation, the entrance of air interfering very seriously with the action of the pump. For this reason the circulating pump is placed as low in the ship as possible, and the bilge suction pipe should be fitted with a non-return valve.

It must not be assumed that all centrifugal pumps lack lifting power. It is not necessary for circulating pumps to have high suction or discharge power, but where necessary centrifugal pumps are designed to discharge water against great pressure, or to raise it to great heights.

The Electric Long Arm System.—The importance of being able to close all water-tight doors with certainty and celerity has caused the introduction of a system by which all doors included in the system may be closed by power from a central or emergency station.

Obviously the requirements of such a system are:

1. That the door may be raised or lowered, by hand or by power, by one operator at the door on either side, and that all doors may be closed simultaneously from one or more emergency stations.
2. That the door may be raised or lowered against a rush of water, or through coal on the door sill due to the first opening of a full bunker.
3. That emergency closing should not interfere with local operation of the door.

4. That while the door is being closed by one operator at an emergency station, it should be possible for a man at the door to stop it temporarily or open it for egress.

5. At the central or emergency stations reliable indicators should show whether the doors are fully closed or not.

A limited number of the most important doors and hatches have been fitted with the electric long arm system in practically all our large ships, though a tendency exists at present to discard the emergency control, and even to discard the local power operation, opening and closing all doors solely by hand power. A few of our older vessels were fitted with practically the same kind of doors, but operated by pneumatic power.

The doors themselves are designed to slide, some vertically and others horizontally, the hatches being raised and lowered on hinges.

Figure 164 shows a vertical sliding power door with electric local and emergency control.

The system here briefly described is the same as that employed with the full-size working models in the Model Room of the Marine Engineering Building, and familiarity with the system will be gained by actually handling these doors during drill periods, rendering an extended description here undesirable.

It is well to point out here, however, some of the important features of such an installation.

The general features of the vertical and horizontal doors are the same. The *bulkhead frame* is a heavy steel casting, securely fastened to the bulkhead and completely surrounding the opening; it forms the foundation for the entire power door. This frame is made sufficiently stiff to prevent any distortion due to irregularity of bulkheads and to secure a perfectly rigid, flat face against which the door plate can seat with uniform tightness.

The *door plate* is made of cast steel, ribbed for combined strength and lightness. The face of the door plate is provided with bronze seating strips to prevent corrosion and preserve good working faces. When unlocked the door plate rides freely in guides with about $\frac{1}{8}$ -inch clearance. *Guides* secured to the bulkhead frame direct the door plate in its movement, and these guides are of open-work construction to prevent clogging by coal, etc., and to facilitate inspection.

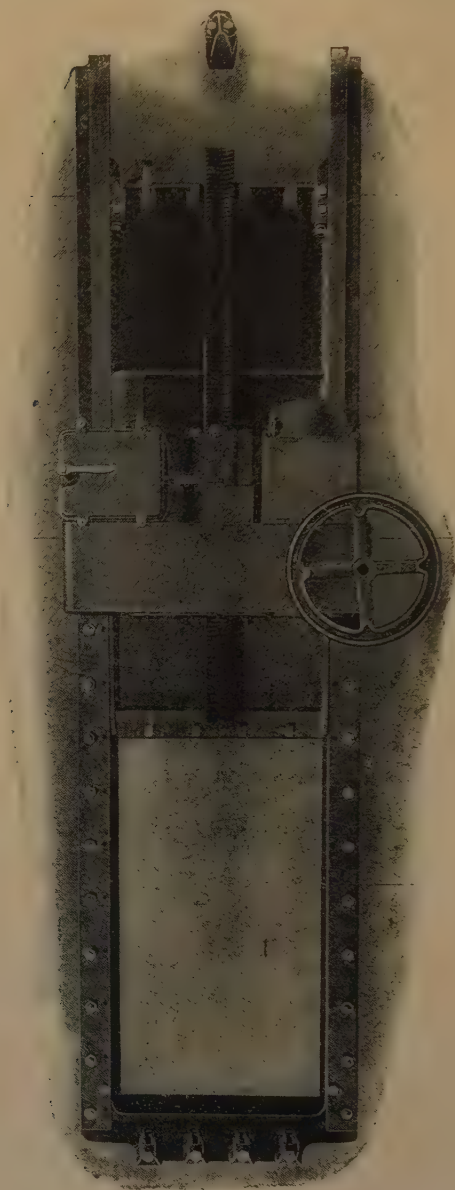


FIG. 164.

The *tightening gear* is important. Each door rides loosely until about to close, when it is driven into its seat by a wedge-shape surface on the closing edge, engaging with rollers secured to the door frame.

There are two important features of the *hatch gear*. First, the *hatch lift*, which is a substantial, rectangular enclosed frame carrying the motor and controller, having on one end a massive projecting crank driven through gearing from either the motor or the hand wheel. The train of gearing is provided with the worm-and-wheel feature, embodying the automatic cut-out device, and by this means the pressure with which the hatch plate is drawn against its seat is a pre-determined and constant amount.

Second, *The lever system for connecting the hatch lift with the hatch plate*, and by means of which the projecting crank on the hatch lift raises, lowers and locks the hatch plate. Secured to the under side of the protective deck and almost directly below the hinges for the hatch plate are two heavy bearings supporting a shaft driven by the crank on the hatch lift. This shaft carries two bent levers extending out beneath the hatch plate and attached to it by suitable links. These levers are placed so that when the hatch is open they occupy but very little of the clear space, being in the corners of the clear opening nearest the hinges.

The links through which the bent levers are attached to the hatch plate serve to operate a tightening and locking mechanism (secured to the under side of hatch plate), which operates after the hatch plate is closed. This locking action is, therefore, performed by the hatch lift after lowering the plate, and does not require any manual operation. The principle of the wedge and roller tightener, used on the power doors, is here applied, and each hatch is firmly secured at its four corners.

The hatch gear is so designed that it may be operated by power or hand, either from the deck or platform below the hatch or from the protective deck.

At each door the following gearing is found:

A hand gear, to open or close door by hand.

A motor with worm gear, pinion and rack.

An electric contact box, for indicating to emergency station when door is closed.

A controller, containing three independent switch mechanisms, viz.: (1) The local-operating switch, by which a man on either side can open and close the door at will. (2) The emergency switch, for closing the door from the emergency station—the action of which may be suspended by the local switch; this will close the door as soon as the handle of the local switch is released. The emergency switch *cannot open* a door. (3) The limit switch, by which the current supplying the motor is cut off when the door has been properly seated or has struck an obstacle with the same force required to close the door tightly. This cut-off is regulated at a certain *power*, not at a certain *position*.

By means of a worm introduced in the motor gearing, an end-thrust of the worm is obtained which bears a certain ratio to the power exerted upon the door plate. This end-thrust is resisted by a helical spring which is at all times subject to an initial compression sufficient to effect a water-tight closure of the door; when the motor power exerted on the door plate exceeds this pre-determined amount the spring yields and actuates the limit switch. It is seen, then, that this switch cuts current off the motor when the force exerted on the door plate exceeds a pre-determined amount which has been found sufficient to secure water-tight closure. And if it be an obstacle which has stopped the door, such as coal, upon yielding of the obstacle the helical spring resumes its normal position and *turns on the current again*.

The Emergency Station consists of a rotary switch mechanism in a water-tight case with outside operating handle. One wire from each limit switch and one from the main line lead into the emergency station box.

Under normal service working, the emergency switch is open. For emergency working, the switch is closed and completes the circuit through the solenoid in the controller box at the door, causing the movable fingers to make contact and complete the motor circuit. As in the case of pneumatic power, during emergency working the above action may be reversed and the door opened by using the controller, as under normal service working. On releasing the controller handle, emergency action is repeated and the door closes and locks automatically.

The Indicator Case at the emergency station consists of the required number of glow lamps. As long as the door is operated only by using the controller at the door, the lamp will not glow, but whenever the doors are closed and locked from the emergency station, each lamp will glow at the moment the door is seated.

The *hand gear* consists of a short shaft extending through the bulkhead and coupled to the extension of the worm and motor shaft. Removable cranks are fitted on each side of the bulkhead and the door opened or closed and locked by hand in case the electric power is not available.

Drainage System.—A brief general description of the drainage system of the battleship *Arkansas* is given below, and Plate 28 shows its main features.

The *main drain*, a 15½-inch pipe, runs from the forward boiler compartment aft on the starboard side to within the engine room, where it is connected to the starboard main circulating pump. A branch goes to the port side and there it traverses the port engine room, with a connection to the main circulating pump aft, and a bilge suction valve in the forward part of the compartment. There is a stop-check suction valve in each engine and boiler compartment as shown.

The suction ends of these valves do not dip into the bilge wells; the valves are placed near the inner bottom, with sufficient clearance above the outer skin not to choke the suction. These main drain valves are operated from 30 inches above the floor plates, and also from the protective deck. Gate valves are placed in the branches leading to the main circulating pumps. Except in case of large leakage that cannot be controlled by the fire and bilge pumps, the main drain suction to circulating pumps are never used.

In each engine room a branch suction from the fire and bilge pump is connected to the main drain cross-connection, at the lowest point of the latter, and is provided with a stop valve and a Macomb strainer. There are no valves or suction in the main drain other than those mentioned above.

The forward *secondary drain*, varying from 4½ inches to 5½ inches in diameter, extends from a manifold at Frame No. 18, just abaft the fresh-water tanks, into and through the boiler rooms.

The after secondary drain extends from the after-trimming tanks into and through the starboard engine room; and a third secondary drain extends fore and aft in the port engine room, with cross-connection to the starboard drain.

These secondary drains have connections to each fire and bilge pump, the after drains to the engine room pumps, and the forward drain to the fire room pumps.

Macomb strainers are fitted in the main lines of the secondary drains to protect all pumps.

Branches from these drains dip into bilge wells in the after end of each boiler room, and in both forward and after ends of each engine room.

There are 3-inch branches from the secondary drains for pumping out all double-bottom compartments inboard of the outermost watertight longitudinal below the armor shelf, except the reserve feed tanks and fuel-oil tanks, and for draining the trimming tanks, drainage tanks, and dynamo condenser compartments. For double bottom compartments in way of engine and boiler spaces the branches are fitted with stop-lift-check valves to admit flooding these compartments through the manifolds from the sea when desired; forward of and abaft the machinery spaces the branches are fitted with stop-check valves, to make it impossible to flood these compartments. Valves in the secondary drains and its branches to bilge wells and double-bottom compartments in way of machinery spaces are operated 30 inches above the floor plates.

Valves controlling suctions forward of and abaft the machinery spaces, except trimming tank suctions and valves in shaft alleys, are operated at the valves and on the berth or protective deck in deck plates.

Valves in shaft alleys are operated at the valves only. Valves controlling the trimming tank suctions are operated on the berth deck.

The chain locker drains into the drainage tank at Frame No. 24.

There are two 6-inch portable hand pumps also supplied, by means of which compartments without fixed drainage may be pumped out.

In general, in addition to the spaces specifically mentioned above as being reached by drainage suctions, the following also have either

fixed drainage pipes connecting with the secondary system, or floor drains leading to compartments reached by the drainage system: Ammunition handling rooms, dynamo rooms, evaporator rooms, ice machine rooms, pump rooms, shaft alleys, steering gear and steering engine rooms, torpedo rooms, turrets, turret foundations, store-rooms and windlass room.

Ammunition rooms, when located on a water-tight platform, usually have no fixed connection to the drainage system.

Compartments not fitted with any fixed drainage are drained when necessary by suction hose temporarily lead as required.

Submerged torpedo tubes have drain pipes leading to the drainage tank.

Such spaces as galleys, pantries, laundries, shower baths, etc., have floor drains discharging into ship's scuppers, or into the bilge if scupper discharge is impracticable. Steam pipes are fitted for boiling out drains from such places as butcher shops, galleys, urinals, etc., and such drains do not lead into the bilge if at all avoidable.

The main drain in early ships ran through the double bottom; this is never the practice now, as injury to the bottom would be apt to injure the drain. The tendency is to localize main drainage by providing separate pumps of large capacity in each of the machinery compartments, and in one recently designed ship there is no main drain. A centrifugal pump in each engine and boiler compartment, driven by motors located on the berth deck, take the place of the main drain when the latter is omitted.

In all cases care is exercised, in running piping, not to impair the water-tight integrity of the bulkheads. Holes through the armor for drainage are avoided where possible.

Flooding System.—All ammunition rooms are fitted with flooding pipes of size sufficient to fill the empty room in twenty minutes. Generally pipes are arranged to flood from the fire main, but sometimes directly from the sea; the latter method was formerly universal, and is now usual in small vessels. Flood pipes end in a sprinkling system designed to spray over all the contents of the magazine, except in projectile rooms, mine-charge rooms, and war-head rooms.

All valves for this system are operated at deck plates on the protective deck. When magazines are flooded air escapes through natural ventilation pipes. There are no drainage pipes to ammunition rooms. Trimming tanks are usually flooded direct from the sea by means of their own sea valves operated from the berth deck.

Many double bottom and other compartments are also arranged for flooding at will. This is accomplished through pump manifolds

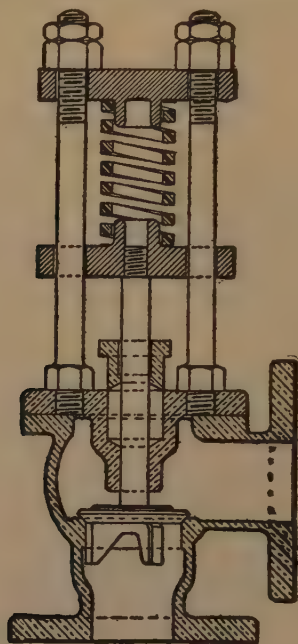


FIG. 165.

by admitting sea water to the proper manifold, then opening the desired compartment valve, which for these compartments is a stop-lift-check valve.

Fire System.—Underneath the protective deck beams the *fire main*, a pipe about 6 inches in diameter in battleships, runs fore and aft through the machinery and magazine spaces on each side of the ship. The two sides are cross-connected forward and aft, forming a complete circuit, to which all fire and bilge and distiller circulating pumps are connected. The cross-connection and cut-out

valves are located so as to permit any portion of the main to be cut out for repairs without preventing use of the rest of the system. Forward and abaft the cross-connections a single main runs to supply the forward and after systems. Horizontal leads *above* the protective deck are avoided when practicable.—

Risers are taken from these mains in number and location such that every point in the vessel except the lower compartments can be reached by a 100-foot length of hose from at least two fire plugs. Branches drop from the main to similarly supply fire plugs for all lower compartments not supplied by risers. All risers have cut-out valves below the protective deck at the main, and others above so located as to localize damage to the system as much as practicable.

Relief valves, Fig. 165, set at 100 lbs., are placed on the fire pump and also on risers forward and aft.

Fire mains usually have connections for magazine flooding, for flushing, for supplying water to torpedo rooms, for air compressors, to cool torpedoes and air flasks, for turret sprinkling, and to flood wing compartments when necessary to correct list in case of damage.

Flushing or Sanitary System.—A flushing main is fitted with branches leading to laundry, washrooms, closets, galleys, general mess pantry, potato peeler, ice machine, circulating pumps, ash chutes and other places convenient for supplying salt water for general use. Direct connection from the distiller circulating pump or fire and bilge pumps, and cross-connection with the fire main supply sea water. A relief valve set at 35 lbs. is fitted to the system.

The crew's lavatory spaces are often flushed by an independent system supplied by electric-motor-driven centrifugal pumps.

Fresh Water System.—In large vessels there are four sets of fresh-water tanks, viz.: *ship's tanks* for storage of drinking water; *reserve feed-water tanks*, for which double bottoms are usually utilized; *gravity tanks*, and *small supply tanks*.

Ship's tanks usually have a capacity of about 30 gallons for each person in the ship's complement. Gravity tanks, located above decks, with about 1-10 the capacity of the storage tanks, afford a constant faucet supply where needed and a means of filling small supply tanks readily; provision is made for filling storage and gravity tanks from a water-boat alongside by pipes with hose con-

nection at ship's side, and from the ship's distilling plant by the distiller fresh-water pump.

A complete fresh-water pumping system (extensive in large ships), operated usually by electric pumps, enables any of the ship's tanks to be filled, and water to be drawn from any tank and discharged into the *fresh-water main*, or into any other of the ship's tanks; supplies water to steamers in their cradles or at the ship's side; to bath rooms, lavatories, pantries, galleys, scuttle butts, bakery, laundry, torpedo rooms, battle dressing-stations, ice plants, instrument sterilizers, and many other places.

The reserve feed-water tanks usually have a capacity of about one ton for each 100 horsepower of main engines.

CHAPTER XIII.

PROPULSION AND SCREW PROPELLERS.

Resistance of Ships.—According to the stream line theory of resistance, when a well-formed ship is towed through water the particles of water are divided at the bow and follow around the side of the ship in well-defined currents and without abrupt or sudden changes of motion. On passing the broadest section of the ship, amidships, the particles close in again over the after part, and leaving the ship form a wake astern. A counter pressure under the stern is caused by the particles thus closing in and tends to balance the head resistance. If the water cannot close in readily and easily owing to the run aft not being fine, an eddying wake will be formed under the stern and decrease the ahead pressure there—equivalent to increasing the head resistance.

Elements of Resistance.—Summarizing the principal facts given by Sir William H. White in the “Manual of Naval Architecture,” regarding the total resistance to the motion of a ship when towed or propelled by sails through the water, and for the present disregarding the resistance due to the action of the propellers in steam ships, the total resistance may be placed under three heads:

1. The frictional resistance due to the rubbing or friction of the particles of water on the skin of the ship.
2. Resistance due to the formation of eddies, known as the eddy-making resistance, due to the wake at the stern.
3. The wave-making resistance.

The frictional resistance is the most important of the above and varies with the area of the immersed surface of the ship; with the condition of this surface, whether foul or clean and smooth; with the length of the surface, and about the square of the speed, and is not sensibly affected by the forms and proportions of ships, unless there is some unwonted singularity of form or want of fairness. This resistance is an important factor at both moderate and

high speeds, for the latter from 45 to 60 per cent of the whole resistance.

The eddy-making resistance is small, except in special cases, a defective form of stern causing largely increased eddy-making. This resistance amounts to about 8 or 10 per cent of the frictional resistance.

The wave-making resistance is the element of the total resistance which is most influenced by the forms and proportions of ships. Its ratio to the frictional resistance, as well as its absolute magnitude, depends upon many circumstances; the most important being the form and length of the entrance and run, in relation to the intended full speed of the ship. For every ship there is a limit of speed beyond which, each small increase in speed is attended by a disproportionate increase in resistance; and this limit is fixed by the length of entrance and run,—the “Wave-Making Features” of a ship.

In addition to the preceding, there is the resistance due to the pitching and tossing in a sea-way, the resistance offered by the air in the passage of the upper body of the ship through that medium, and the propeller “augment” of resistance.

Augment of Resistance due to the Propeller.—This is also known as the “thrust deduction” and arises from the suction of the water from the stern of the ship. The water displaced at the bow instead of closing in and around the stern according to stream lines in a well-formed ship, is more or less disturbed and drawn away from the stern to the propeller. In consequence, the pressure of water under the stern is decreased, and increases the resistance of the ship as compared with its natural resistance, as for instance, when being towed.

This augmentation varies considerably and its value depends chiefly upon the form of the stern, the speed and the dimensions, number and position of the screw propellers. From the experiments conducted by Mr. Froude, Mr. Yarrow and others, the augmentation for ships with thick stern posts, full forms and single screws, a value of 40 per cent was given, the rudder post representing about 10 per cent. Subsequent experiments gave from 20 to 40 per cent in single screw ships, the lower value corresponding to

the finer forms. It was also shown by Mr. Froude's experiments that if the single screw was placed about 30 per cent of the extreme breadth of the ship clear of the stern, the augmentation was only one-fifth as great as when it occupied its usual position. While such a change of position reduces augmentation, it lessens the utilization by the propeller of the energy in the following wake of water, so that the gain in one direction is neutralized by a loss in the other. For twin-screw ships the augmentation varies from 25 per cent in full ships to 6 per cent in fine ships.

Coefficients of Performance.—In calculating the horse power required to drive a projected ship at a given speed, an approximation is made by using coefficients based upon the recorded trial performance of preceding ships. Among the various formulae used in powering ships, the Admiralty coefficients, to indicate the approximate resistance and propulsive efficiency, were formerly widely used for vessels of similar form and proportion, and not far from the same size:

$$C_1 = \frac{A V^3}{\text{I.H.P.}} \quad (1)$$

$$C_2 = \frac{D^2 V^3}{\text{I.H.P.}} \quad (2)$$

D = Displacement of ship in tons,

A = Immersed midship section in square feet,

V = Speed in knots per hour.

These formulae are based upon the assumptions: (1) that the resistance to the motion of the vessel varies as the square of the speed, and consequently the work required to overcome this resistance varies as the cube of the speed; (2) that the indicated horse power varies as the effective horse power; (3) that for similar ships, the resistance corresponding to any given speed will vary as the immersed midship section, or the displacement to the two-thirds power.

The first assumption is only true for low speeds, as the resistance at the higher speeds varies at a higher power than the square, while the second assumption is no longer true when the speed brings in wave-making resistance, and the third assumption is only true for a limited range of power in an engine. These coefficients although

extensively used before the era of high powers and wide variation in the types of ships, are not sufficiently accurate for modern use. Of the two coefficients that for the displacement is the more trustworthy.

In using these formulae great care should be taken in the proper selection of the value of C , the usual method being to find values of C from the performance on speed trials of actual ships of the same size, character, form and speed. When used discriminately these formulae can be made of much valuable assistance to the designer in determining power or speed for a new design, but must be used with caution and their limitations well understood.

These coefficients are not constant numbers, but are different for different ships and vary with the speed even in the same ship.

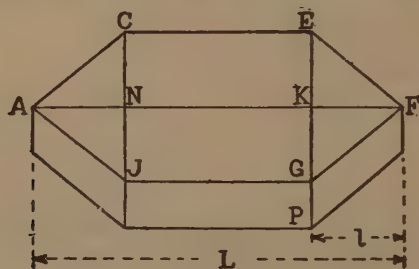


FIG. 166.

Kirk's Analysis.—A method of analyzing the resistance and indicated horse power based upon the wetted surface itself, or an approximation was devised by Dr. Kirk.

In this method the hull below the water line is represented by a block model, Fig. 166, in which the middle body is a rectangular parallelepiped and the fore-body and after-body, equal prisms with isosceles triangles for bases.

Let L equal the length of a ship, B the beam, H the mean draft, D the displacement in tons and M the area of the immersed midship section.

In Fig. 166, the block model is supposed to float level with the surface of the water, and has the same length, a draft equal to the mean draft of the ship, and beam B' such that B' multiplied by the mean draft is equal to the immersed midship section.

Since the triangles at the ends have equal bases and altitudes, they together will equal the rectangle $CJ \times AN$, hence the area $ACEFGJ = EG \times AK$. The total volume of the block equals $EG \times AK \times GP$.

$EG \times GP$ is equal to the I. M. S. of block, and is equal to the I. M. S. of ship (by supposition). Since volume displaced by the block model equals the volume displaced by the ship.

$$\text{Displacement} \times 35 = \text{I.M.S.} \times AK$$

$$\therefore AK = \frac{D \times 35}{M}$$

$$KF = L - AK.$$

$$EG = M \div \text{mean draft.}$$

$$\text{Area of bottom} = EG \times AK = B'(L - l) = \frac{M}{H} \times \frac{35D}{M} = \frac{35D}{H}$$

$$\text{Area of sides} = 2 \times JG \times GP = 2H(L - 2l)$$

$$\text{Area of ends} = 4H \sqrt{\left(\frac{B'}{2}\right)^2 + l^2}$$

The angle of entrance may be obtained from tangent of $\frac{1}{2}$ angle of entrance, $EFK = \frac{B' \div 2}{l}$.

Having approximately obtained the wetted surface, Dr. Kirk assumed that resistance varied directly as the surface and as the square of the speed, and that the indicated horse power would vary as the cube of the speed, and employed the following formula:

$$\text{I.H.P.} = K \left(\frac{S}{100} \right) \times \left(\frac{V}{10} \right)^3$$

in which K horse power is assumed per 100 sq. ft. to drive the vessel at the rate of ten knots per hour, and that the quantity varies as the cube of the speed. The values given by Kirk for K are, 4 for fine ships with clean bottom and high propulsive efficiency; 5 for merchant ships of ordinary propulsion and efficiency; and 6 for short broad ships. This formula was intended to be applied to vessels of moderate speeds. For high speeds this method is used by considering that in passing from ten knots to the high speeds, the I. H. P. required will vary as the ratio of the speed raised to the 3.5 power.

This method has the advantage over the Admiralty coefficient

method, in using a close approximation to the wetted surface itself, instead of ratios varying with the ships, but like the Admiralty coefficient method it is based upon assumptions which may be close approximations at low speeds and not true at high speeds.

In the application of Kirk's method to a projected new ship, the values given the constant K should be deduced from well established trial results of ships similar in type and speed, to the proposed ship.

Froude's Law of Comparison.—By the term "Corresponding Speeds" for similar ships is meant speeds proportional to the square root of their linear dimensions. For instance, a ship 400 feet long is to be driven at 16 knots per hour, the corresponding speed on an exactly similar model ship 16 feet long would be in the ratio of

$$\frac{\sqrt{16}}{\sqrt{400}} = \frac{4}{20} = \frac{1}{5} = 3.2 \text{ knots.}$$

Briefly Froude's law of comparison is that the skin resistance of a ship and its model, or of two ships of different sizes but of the same form, varies in the ratio of their displacements at corresponding speeds, or,

If the linear dimensions of a vessel be L times the dimensions of the model, and the resistance of the latter at speeds V_1, V_2, V_3 , are R_1, R_2, R_3 , then at the "corresponding speeds" of the ship $V_1\sqrt{L}, V_2\sqrt{L}, V_3\sqrt{L}$, the resistance will be R_1L^3, R_2L^3, R_3L^3 .

This law of comparison for the resistance of a ship or model can be readily applied to calculating the necessary power.

Let R, V, D, R_1, V_1, D_1 , represent the resistance, corresponding speeds and displacement of two similar ships.

$$\text{Then } \frac{R}{R_1} = \frac{D}{D_1}, \text{ and } \frac{V}{V_1} = \sqrt{\frac{L}{L_1}} = \sqrt{\left(\frac{D}{D_1}\right)^{\frac{2}{3}}} = \left(\frac{D}{D_1}\right)^{\frac{1}{3}}$$

$$\text{Hence, } \frac{RV}{R_1V_1} = \frac{D}{D_1} \times \left(\frac{D}{D_1}\right)^{\frac{1}{3}} = \left(\frac{D}{D_1}\right)^{\frac{4}{3}}$$

R, V and R_1, V_1 are proportional to E and E_1 , the effective horse power, so that

$$\frac{E}{E_1} = \frac{RV}{R_1V_1} = \left(\frac{D}{D_1}\right)^{\frac{4}{3}}.$$

If the efficiency of machinery, propellers, etc., is the same in both cases, the ratios of effective to indicated horse power will be the same, so that

$$\frac{E}{E_1} = \frac{\text{I.H.P.}}{\text{I.H.P.}_1} \text{ and } \frac{\text{I.H.P.}}{\text{I.H.P.}_1} = \left(\frac{D}{D_1}\right)^{\frac{7}{6}} \text{ or } \text{I.H.P.}_1 = \text{I.H.P.} \times \left(\frac{D_1}{D}\right)^{\frac{7}{6}}$$

$$V_1 = \left(\frac{D_1}{D}\right)^{\frac{1}{6}} V.$$

Having given the I. H. P., V and D for a given ship or model, the indicated horse power (I.H.P.) and corresponding speed (V_1) for a similar ship of given displacement can be readily obtained. Or, from a speed curve of a ship of known dimensions, the corresponding curve.

Effective Horse Power.—If a ship is towed at a uniform speed by some external force which does not interfere with the flow of water past her hull, the resistance experienced by the ship is measured by the strain on the tow rope. Calling this resistance R and the speed S feet per minute, the work done or the effective horse power is,

$$\frac{R \times S}{33,000}.$$

When the ship instead of being towed is driven by her own propeller and machinery the resistance will be greater than the tow boat strain for the same speed, owing to the additional resistance introduced by the propeller, frictional, thrust deduction, etc., previously described, and frequently the propeller produces motions and reactions of the water in direction not effective for propulsion. Indicated horse power is the work done by the steam in the cylinders of the engine, and between the indicated power exerted by the engine and delivered to the propeller there is considerable waste due to frictional and other resistance of the mechanism.

Curves of Performance.—The method of expressing the results of a progressive trial graphically is as follows: Diagrams are constructed in which the speed in knots per hour are set out as ordinates and the corresponding rev. per min. as abscissae. In

Fig. 167 the curves of R.P.M.—speed, and R.P.M.—shaft horse power, are shown for the U. S. S. Wyoming.

The Value of Performance Curves.—From an examination of performance curves, the following can be obtained:

1. The indicated horse power, revolutions and slip corresponding to any intermediate speed between observed speeds.

2. The general engine efficiency and comparison with other engines.

3. The rate of increase of power for speed, and if the resistance is fairly following the laws of resistance.

4. The defects in the propeller observed from the slip curves.

5. The power required to drive a similar ship in size and shape at a greater or less speed from an inspection of the curves of I.H.P. and coefficients, and the power required for approximately similar forms when the size and speed are both varied, by the application of Froude's law as previously explained.

Mr. Froude proposed the substitution in speed curves, of indicated thrust in place of horse power, because the latter includes in one term the merits of the ship, engine and propeller, and because the term into which it groups these items is complicated by the introduction of the speed factor, instead of representing them under a force factor. With this view he converts the horse power term into a force term, by dividing it by the speed of the propeller. The result he terms the *indicated thrust* and it is the thrust which the propeller would be exerting if the force of the steam were wholly employed in creating thrust, instead of overcoming friction and other collateral resistance.

$$\text{Indicated Thrust} = \frac{\text{I.H.P.} \times 33,000}{\text{pitch} \times \text{revolutions}} = \frac{2 P L A N}{\text{pitch} \times N} = \frac{2 P \times L \times A}{\text{pitch}}$$

This thrust corresponds to an absence of slip and a total efficiency of unity, an impossible condition, and is only used in expressing certain relations in ship's performance.

In laying down the curve the calculated indicated thrust from the trial speed is set off as ordinates on a base line of speeds.

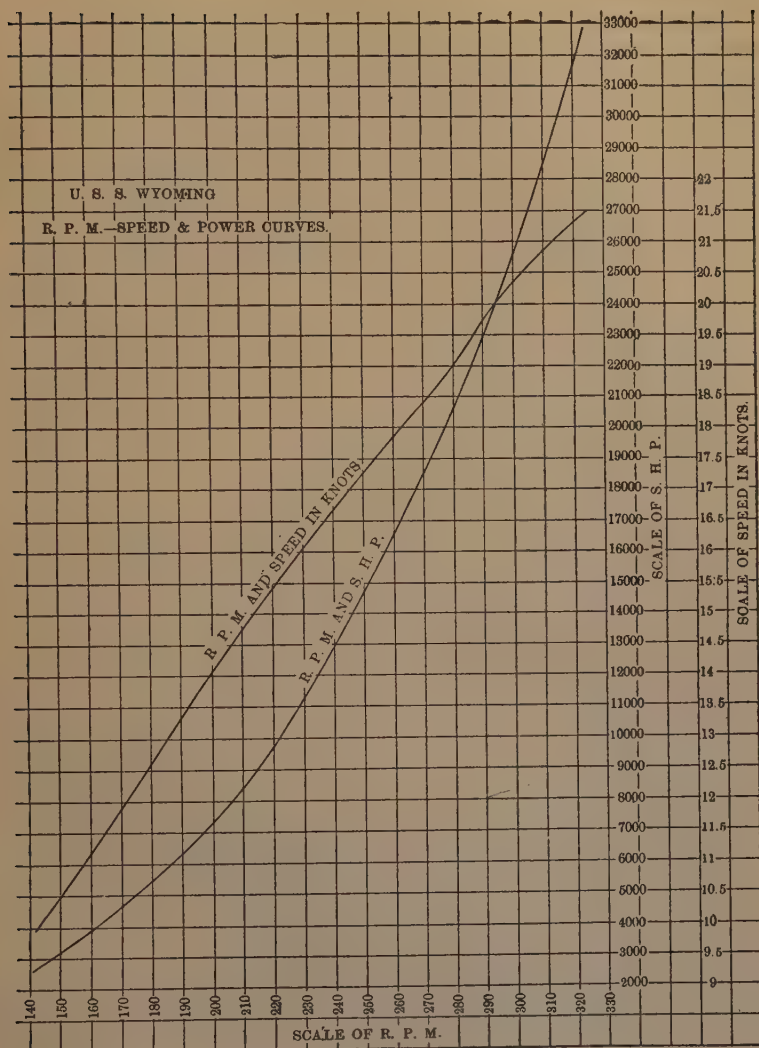


FIG. 167.

In drawing the indicated thrust curves from the data of actual trials, Mr. Froude observed that at the low speed end, the curve did not descend to the thrust zero, but crossed the vertical axis at a point representing a considerable amount of thrust. This apparent thrust at the zero of speed when there can be no real thrust is the equivalent of the initial friction. If a horizontal line be drawn through the intersection of the curve with the axis of Y and parallel to the axis of X , the height thus cut off from the thrust ordinates would represent the reductions to be made from them for the constant or initial friction, and the remainder would be approximately proportional to the ship's true resistance.

To fix this point of intersection, Mr. Froude offered the following method: The lowest point of the curve is fixed by the data given at 3, 4 or 5 knots. At these speeds, the resistance has been found to vary approximately as the 1.87 power of the speed. Hence, on this assumption the lower end of the thrust curve when divested of the constant friction should be a parabola in which the ordinate is as the 1.87 power of the abscissa.

In Fig. 168 let $A-B-C$ represent the lowest portion of the thrust curve. From the point A near the lowest point of the curve draw the tangent PP' ; draw the vertical at H' so as to cut the space OH into segments the ratio indicated 0.87 to 1. Draw a line parallel to OX through the point where the vertical H' cuts the tangent; the point D where this line cuts the vertical axis will be a point through which the curve will pass and OD will represent the thrust equivalent to the constant or initial friction.

From numerous experiments this method cannot be relied on for giving accurate results, from the variations in the estimates of initial friction thus obtained.

Propulsive Coefficient.—The ratio which the effective horse power bears to the indicated horse power for a given speed is known as the *propulsive coefficient*.

The values of propulsive coefficients have been obtained from numerous experiments on different types of ships, and as given by Sir William White, are as follows:

Single screw ships, thick wooden stern posts.....	40
Steel stern posts and fine forms of stern.....	50 to 55
Twin screw ships of fine form.....	50 to 60
Torpedo boats.....	55 to 65

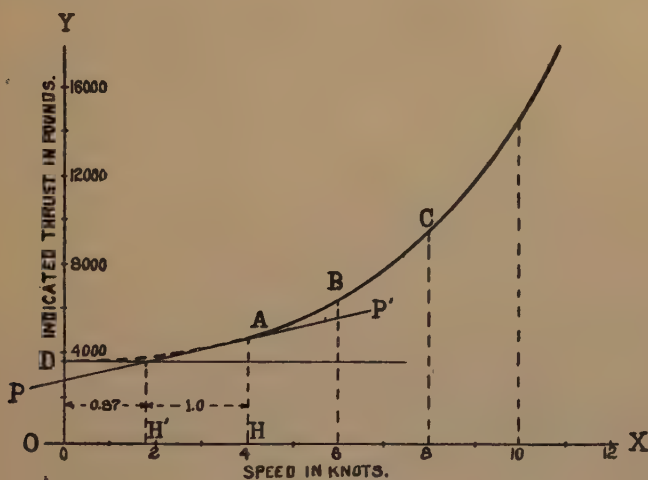


FIG. 168.

Distribution of Power for Propulsion.—From the above it is seen that in a well designed ship the indicated horse power is nearly twice the effective horse power. An examination into the cause of this waste of work shows it to be due to frictional resistances. (1) Owing to the tightness of bearings, pistons, valves, etc., a definite amount of power is required to start the engine against its own resistance; this is known as *initial* or *dead load* friction, and is practically constant for all speeds of revolution. (2) The friction due to the working load and dependent upon the thrust of propeller and speed of engines and ship. (3) The waste work due to the action of the propeller.

The distribution of power with a modern high speed engine with forced lubrication is about as follows:

Initial friction	.04
Working load friction.....	.04
Friction and resistance of propeller, thrust deduction and allowing for wake gain.....	.27 to .35
Effective horse power.....	.65 to .57
Total	1.00

The first two items represent the loss due to the mechanism or engine losses, giving an engine efficiency of .92, or 92 per cent of the power is delivered to the screw. At full power this friction loss in engines not fitted with forced lubrication may amount to a mean value of 15 per cent, giving as the engine efficiency 85 per cent.

At reduced speeds the friction is reduced at a much lower proportional rate than the power. Taking the friction at full power as given above, 8 per cent, and calling the power 100, there would be required for one-half speed, about one-eighth power or about 12.5. The friction of load reduces in the same ratio, or $4 \div 8 = .50$, while the initial friction reduces directly as the revolutions, to half its full power value approximately, or 2.0. The total friction loss is now 2.50 for a power of 12.5, or 20 per cent of the power required for the reduced speed. This illustrates the wastefulness at the reduced power of naval vessels, made necessary for cruising conditions, and the difficulty in obtaining good propulsive efficiencies under the condition of very high powered machinery running at one-eighth or one-tenth the designed power.

The distribution of power may be shown graphically on a diagram by plotting the total power and the different frictional components on speed as abscissae. Fig. 169 shows such a diagram giving the distribution of the Yorktown's power as plotted by Naval Constructor Taylor in "Resistance of Ships." Ordinates of the curve 1 show the horse power absorbed by the initial friction. Ordinates

between 2 and 4 show the loss by friction and resistance of propeller. The wake gain and thrust deductions are nearly identical, the former being represented by the ordinates between 4 and 3, and the latter by the ordinates between 3 and 4. Between 4 and 5 the ordinates show the power absorbed by skin resistance and between

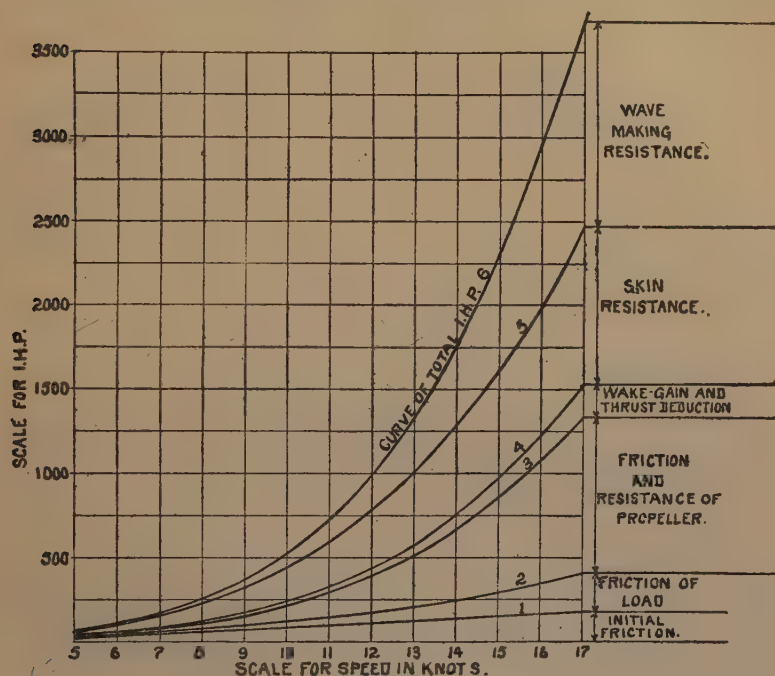


FIG. 169.

5 and 6, the wave resistance power. The total indicated horse power is given by ordinates from the base to the curve 6.

The application of curves as mentioned in the preceding can be carried out in cruising and can be made the source of much valuable information in reference to the performance and efficiency of machinery on board ship. Naval vessels in service use a range from the lowest speeds up to the maximum, under natural and

forced draught, while indicator diagrams are taken daily. This, together with the detailed data furnished by the steam and navigation log books gives a complete history of the conditions, and when tabulated and shown in the form of a curve furnishes a ready method at hand of obtaining much useful information, without laboriously overhauling and computing data.

Plate 29, showing the results of the official progressive speed trials of the U. S. S. Cleveland, and tabulated by Lieut.-Commander Bowers, U. S. N., illustrates the use of curves of performance and shows the ready and concise forms in which all data and resulting values obtained by systematic observation may be plotted for reference and comparison. If merely tabulated they would require much time and difficulty in unraveling.

Power—Speed—Coal Consumption.—The effective power has been given as $\frac{RV}{33,000}$, and if it is assumed that the effective power is

a definite ratio of the indicated power, then since R varies as V^2 , the power may be said to vary as $R \times V$, or,

$$\text{I.H.P.} \propto V^3 \quad (1)$$

This is only true for moderate speeds for the particular ships under discussion, for instance, up to within 3 to 4 knots of the full speed of ships designed for 15 to 20 knots. Beyond this point the power increases from the cube of the speed up to the fourth and fifth power for the last knot gained, and depends greatly upon the shape of the vessel and the speed at which she is forced. Also, at very low powers, the horse power does not decrease as rapidly as given above, from the fact that the dead load or constant friction remains nearly the same; also, the cost of auxiliaries and their effect becomes very disproportionate at low speeds.

On the same assumption,

$$\text{Coal consumption per hour} \propto V^3 \quad (2)$$

Dividing through by the speed,

$$\frac{\text{consumption per hour}}{\text{speed per hour in knots}} = V^2$$

$$\therefore \text{Consumption per knot} \propto V^2 \quad (3)$$

Multiplying through by distance run at a given speed V :—

$$\text{Consumption for a known distance at a given speed } (v) \propto V^2 \times \text{Distance.}$$

The above general laws enable comparisons to be made between coal consumption for the results achieved; they are of limited application, for they are largely modified by the following:

1. The condition of the ship's bottom, whether clean or foul.
2. The wind and weather conditions.
3. Quality of the fuel employed.
4. The condition of the machinery.
5. The number of auxiliaries in use, and appliances consuming steam not employed in driving the ship.
6. The design of the hull and machinery.

With the exception of the last, the above factors are all very variable and erratic. A curve constructed from the data in the log book is apt to be very misleading, and frequently produces very discordant results. The speed entered in the steam log is that of the patent log without subsequent corrections made by the navigator at the end of the run, while the force of the wind and state of the sea are apt to be very vague as far as determining their effect upon the ship. The coal consumption as entered upon the steam log needs frequent checking with the contents of the bunkers, to see that the amounts turned in are correct. As shown below, the number of auxiliaries used, other than that necessary for propulsion, such as apparatus for ventilating, heating, lighting, distilling, etc., are exceptionally important in estimating the radius of steaming action.

Under these conditions a coal consumption curve obtained by simply taking the data from the log and sweeping a fair curve through the middle of the series of points obtained will give results of not much benefit, unless great care is taken in obtaining correct data for entry in the steam log.

A *coal consumption curve* is made by making a number of runs at various speeds and carefully measuring the coal used. The various points ascertained are set off on a diagram and a *fair curve* drawn through them. As shown in Fig. 170, this curve like the curve of indicated thrust does not pass through the origin. For any intermediate speeds between the points observed a simple measurement of the diagram will show the consumption required per day. The most economical speed can also be found graphically from the diagram by drawing a tangent to the curve from O meeting it at A .

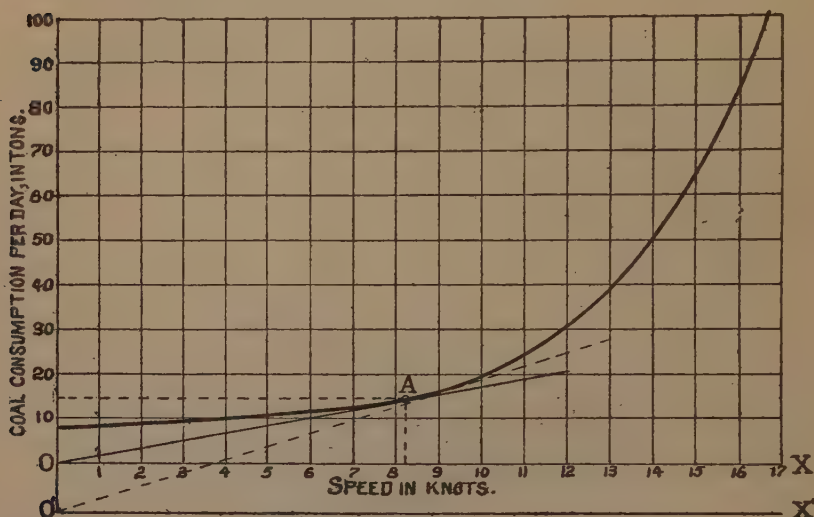


FIG. 170.

The ratio of coal consumption, that is, the tangent of the angle AOX is least at the point A and is greater at any other point of the curve, hence, the point given by A will be the most economical speed of the ship.

In estimating the steaming radius of a ship, the amount of coal used for other purposes than steaming with the main engines should be taken into consideration. To do this lay off $O'X'$ at a distance from OX equal to the consumption for distilling, electric lighting and other auxiliary purposes about the ship, and from the

point O' draw a line tangent to the curve, the point of tangency as before giving the desired economical speed, or minimum value of the ratio $\frac{\text{Total Expenditure}}{\text{Speed}}$.

Coal Consumption for Auxiliary Purposes.—The coal consumption on board naval vessels for auxiliary purposes, other than what is needed for propulsion, such as heating ship, distilling water for ship's use, electric lights and motors, refrigerating plant, etc., amounts to a considerable item in the coal consumption of a ship. In modern naval vessels in commission, the following are the amounts consumed for auxiliary purposes for port service: Large first-class battleships and large armored cruisers, from 15 to 30 tons, protected cruisers, from 15 to 20 tons, and gunboats, from 3 to 6 tons.

Instead of passing the exhaust steam of the auxiliaries into the auxiliary condenser and thus carrying off the heat contained in the exhaust to waste, the auxiliary exhaust pipe is connected to the evaporator coils and the auxiliary condenser cut out. By this means the heat of the exhaust is utilized in the production of fresh water and the cost of working the auxiliaries reduced. Relief valves are fitted so that if the exhaust is in excess of the requirements, communication is opened to the auxiliary condenser in port, or to the second receiver when the ship is underway.

The most economical speed and the relation of coal expended for auxiliaries underway will be referred to separately, but, in passing, the great proportion of the coal expended on auxiliaries, in comparison with that on the propelling engines alone at low powers is shown in an article on the coal consumption of the *Charleston* by Professor Hollis (formerly Passed Assistant Engineer, U. S. N.), before the American Society of Naval Engineers.

From the approximate estimates of the I. H. P. and coal in the accompanying table from the article mentioned, it will be seen that of the total coal expended 74.2 per cent is expended on auxiliaries and losses alone at 4 knots, 46.6 per cent, at 8 knots, and 15.8 per cent at 18 knots. These results show the reason for the great falling off in economy in the main engines of naval vessels when working at low powers.

TABLE VII.

Speed in knots.	I. H. P. main engine.	Coal per I. H. P.	I. H. P. Total auxiliaries.	Coal per I. H. P. Auxiliaries.	Coal per hour, auxiliaries.	Coal for radiation, leakages, etc.	Coal for distil- ling.	Coal for main engines.	Total coal per hour.	Knots per ton of coal.	Per cent coal on main engines.
4	107	4.0	79.	6.	474	655	100	428	1657	5.41	25.8
6	270	3.25	81.5	6	489	655	100	878	2122	6.33	41.4
8	520	2.8	85.9	6	515	655	100	1456	2726	6.57	53.4
10	960	2.5	93.4	6	560	780	100	2400	3840	5.83	62.5
12	1710	2.3	107.6	6	646	905	100	3933	5584	4.82	70.4
14	2820	2.1	130.1	6	781	1030	100	5922	7833	4.00	75.6
16	4370	1.9	179.5	6	1077	1030	100	8303	10510	3.41	79.0
18	6120	2.5	288.7	6	1732	1030	100	15300	18162	2.22	84.2

In these estimates Prof. Hollis assumed a coal consumption of 6 pounds per I. H. P. per hour for auxiliaries. Subsequent experiments have demonstrated that the actual consumption was even in excess of this amount. In a later and elaborate series of experiments on the auxiliaries of the U. S. S. Minneapolis by Lieut.-Commander W. W. White, the startling figures were shown of engines with a steam consumption of from 100 to 300 pounds per hour per I. H. P. Of the 34 auxiliaries tested on the Minneapolis, 31 were indicated giving an aggregate of 471 I. H. P. and an average consumption per hour of 120 pounds of water per I. H. P., while the main engines developing less than one-twelfth, their maximum horse power gave an average economy of 21 pounds per I. H. P. At eight pounds, evaporated per pound of coal, the average cost in coal per I. H. P. per hour of the auxiliaries would amount to 9.6 pounds.

The Screw Propeller.—The fundamental principle upon which nearly all marine propellers act, is that the reaction due to projecting a mass of water in one direction produces motion of the ship in an opposite direction. The sternward momentum communicated in each unit of time by the propeller to the water on which it acts, measures the thrust of the propeller.

If V represents the velocity of the stream of water with reference to the ship, and v the velocity of the ship, $V-v$ will represent the velocity of the stream over the water on which it is projected.

The propelling effect is equal to the mass projected in a given time multiplied by its velocity. If M is the mass of water projected in a second and $V-v$ its velocity in feet per second, then, the momentum $= M(V-v) = \frac{W}{g}(V-v)$.

The weight of water projected per second, if A represents the area in square feet of the projected stream, is $A \times V \times 64$, (taking the weight of sea water at 64 pounds per cubic foot).

$$\text{Momentum of Stream} = \frac{A \times V \times 64}{32}(V-v) = 2A \times V(V-v).$$

From the principle given above, this equals the reaction produced, or the thrust of the propeller in pounds. In a screw propeller the velocity (V) of the stream sent aft would be given by the product of the pitch and the revolutions per second, while the area (A) of the section of the stream is approximately the area of the circle

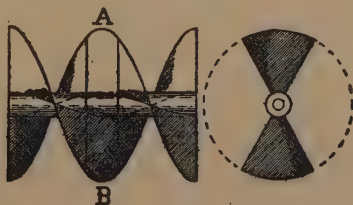


FIG. 171.

of the same diameter as the screw, minus the area of the transverse section of the hub.

While the above is the theory, were the conditions simply that of driving a column of water astern and obtaining the thrust due to the reaction, the action of the screw propeller modifies the conditions and is complex, from the following causes: Instead of a column of water of a circular cross section (or race) moving aft with a direct sternward acceleration after passing through the propeller, it is also given a rotary motion. This is accompanied by a centrifugal force which diminishes the pressure from the center of the column to outer surface. Also, the screw instead of acting upon undisturbed water, acts upon water which has already been set in motion by the advance of the ship.

The difficulties attending a complete investigation of the effi-

ciency of the screw propeller depend upon so many factors that although much thought and investigation have been brought to bear on the problem it is yet unsolved, and designing engineers are largely governed by the results obtained on trial investigations and model tank experiments.

In the discussion of, and practical dealings with the screw propeller, the following technical terms are in constant use.

If a flat plate be wound edgewise spirally round a cylinder and then a slice, *AB*, be cut off as in Fig. 171, there will be one piece of the blade attached. If two plates are used, forming a double-threaded screw, or three plates forming a triple-threaded screw, then the section cut out will give two blades or three blades, and the propeller a two-bladed or three-bladed screw.

Three-bladed screws are in general use in the naval service. They are more steady in their action in a sea-way, produce less vibration and are less affected by the loss of a blade than two bladed screws.

Length of Screw.—As stated above, the screw blade is only a portion of a complete convolution, the length of the screw is the extreme length of the blade measured parallel to the axis of the shaft.

Diameter.—The diameter of the screw is the diameter of the circle described by the tips of the blades.

Pitch.—The pitch is the length of a complete turn of a blade or thread measured parallel to the axis. This is only true when the pitch is uniform over the surface of the blade. Screw propellers are generally constructed with a varying pitch, so that this definition applies to but one element of the driving face, and is the distance the ship would move if this element were working in a fixed nut.

Uniform Pitch.—The pitch is said to be uniform when its value is the same at all points of the surface of the blade.

Varying Pitch.—If the pitch increases from the center towards the periphery, the blade is said to have a radial expanding pitch. If it expands along the axis from the entering to the leaving edge, the blade has an axially expanding pitch. In these cases the

“mean” pitch is ascertained by measurement, and is used in all calculations of performance.

Fraction of Pitch.—The ratio of the length of the blade to the pitch is termed the fraction of the pitch in comparing different screws.

Angle of the Screw.—If AB , Fig. 172, represents the axis of the screw and a line, AC , perpendicular to the axis, travels uniformly around it, and at the same time along the axis, the surface swept up will be a helical or screw surface, the angle CDE between the curved surface and horizontal plane BED is called the angle of the screw at the radius AC . With a uniform pitch, the angle

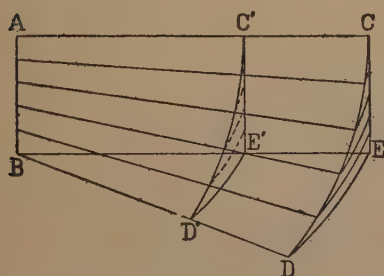


FIG. 172.

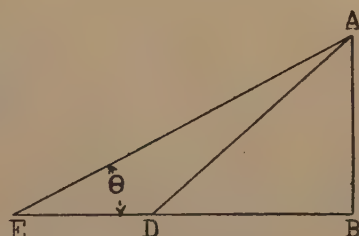


FIG. 173.

increases as the radius decreases as shown at the radius AC' , the angle $C'D'E'$ being much greater than CDE .

The relation between the pitch, circumference and angle of the screw may be shown by a right-angled triangle set off with a base equal to the circumference of the screw and the altitude equal to the pitch. In Fig. 173, let BE drawn to scale represent the circumference described by the extremities of the blades and AB the pitch, then the angle of the blade at the tip is given by,

$$\tan \theta = \frac{\text{pitch}}{\text{circumference}}.$$

Similarly, the angle at any other point, assuming the pitch the same throughout, may be found by laying off the circumference due to that radial distance from B , such as D , then BDA gives the angle of the blade for the point measured.

Face of the Blade.—The rear face is termed the face or driving face of the blade and drives the water aft when going ahead.

Back of the Blade is the opposite face and consequently on the forward side.

Leading Edge.—The edge which cuts the water first when the screw is turning ahead is called the leading edge.

Following or Leaving Edge is opposite the leading edge.

Propeller Hub.—The hub or boss is the spherical or cylindrical center to which the propeller blades are attached.

Right Hand Propeller.—A screw propeller is termed right or left hand when it turns with or against the hands of a watch when looked at from aft and the ship in the ahead motion.

Propeller Race.—The cylindrical column of water driven aft by the action of the screw propeller is called the propeller race. The cross sectional area of this race is measured by the area of the screw's disc minus that of the hub.

Pitch Ratio.—The pitch ratio is the ratio of $\frac{\text{pitch}}{\text{diameter}}$.

Disc Area.—The area of the circle described by the tips of the blades is termed the disc area.

Helicoidal Area.—The actual surface of the face of a blade is the helicoidal or developed area of the blade and for the whole propeller is the sum of the helicoidal areas of the blades.

Projected Area.—The area of the projection of one blade or all blades on a transverse plane is known as the projected area.

Speed of Screw.—By the speed of screw propeller is meant the distance it would advance in a unit of time supposing there is no slip or the screw working in a solid nut. The speed of the screw is equal to the pitch multiplied by the revolutions made in a unit of time.

Slip.—Instead of the screw propeller working in an unyielding surface, it moves in a yielding medium, water, and the advance is less than the pitch multiplied by the revolutions. The difference

between the speed of the screw and the speed of the ship is called the apparent slip of the screw.

Let V = the speed of the ship,

v = the speed of the screw,

$v - V$ = apparent slip of the screw.

$$\text{Then, } \frac{v - V}{v} \times 100 = \text{percentage of slip.} \quad (1)$$

The apparent slip is determined on the assumption that the water on which the propeller is acting was previously at rest. It has been explained that a ship in motion gives a forward movement to the water immediately surrounding the hull, nearly equal to its own velocity at the surface of the hull, and rapidly decreasing away from the surface. This frictional movement of the water together with the wave and stream line motion concentrates at the stern and forms the "wake," with a forward velocity.

The "true slip" then should be that given by taking the speed relative to the "wake" of water in which it is acting, while the "apparent slip" is that given by taking the speed relative to the outside still water.

Assuming v_0 to be the speed of the wake, if the propeller were not acting and the ship towed at the speed V , we have,

$$V - v_0 = \text{speed of advance of propeller relative to wake, and the true slip equals } \frac{v - (V - v_0)}{v} = \frac{v + v_0 - V}{v}. \quad (2)$$

The speed of the wake is difficult to obtain and varies with the form of the ship, so that in general when the slip is mentioned the apparent slip is meant, and not the real slip, unless the contrary is stated.

The loss from slip is a necessity, for in order to obtain propulsive thrust, a body of water must be driven astern with an increased velocity, and the water must yield under the action of the propeller.

With an efficient propeller, the slip varies from 10 to 20 per cent, and may be much less. With a very foul bottom or a poor form, the velocity of the wake may become very great, and one part of the total slip may become very large, and the other part, the apparent slip, correspondingly small, but the total or real slip, the

sum of the two will still be high and is the definite quantity dependent upon the efficiency.

If the slip is very small an examination will show an excessive amount of power employed, for the forward current is produced by the engine power and this increased amount required, is greater than ordinary.

Negative Apparent Slip.—The above reasoning will account for the condition of negative slip, for if the initial velocity of the wake be sufficiently great the apparent slip may become negative, that is, the speed of the ship might be greater than the screw. This condition can be readily seen from formula (2), for if v_0 is very large the speed of the ship V could consistently be greater than v and the apparent slip would be negative, but, with a positive value for the real slip.

In some cases apparent negative slip is ascribed to the errors in estimating the true mean pitch of a screw, the usual method being to measure the pitch at various equidistant points and taking the arithmetical mean for all blades, a method that is not wholly certain of determining the true mean pitch as far as the propulsive effect is concerned. Regardless of this, however, apparent negative slip is always possible with a wake speed greater than the speed of the ship. This wake can be attained in large, full stern ships when a large stream line wake is added to the frictional wake. Negative slip is always a sign of low true slip and consequently of low propeller efficiency, while the ship resistance must be excessive to correspond to the formation of such a wake velocity.

General Conclusions.—The form of the after part of the ship should be such as to permit the free flow of water to the screw.

Proper Immersion.—The blades should not break the surface of the water or draw down air to decrease the quantity of water acted upon by the propeller.

There is a considerable waste of power in overcoming the edge-wise or frictional resistance offered by the water, to a rapidly revolving screw propeller.

Normal Thrust.—The mean normal thrust along the shaft of an engine may be obtained as follows:

Let T = mean normal thrust exerted,

S = speed of vessel in knots per hour,

then, the work done in a unit of time is $T \times S \times \frac{6080}{60}$

The effective horse power equals $\frac{T \times S \times 6080}{60} \div 33,000$.

Assuming E.H.P. = $\frac{2}{3}$ I.H.P.,

$$\text{I.H.P.} = \frac{T \times S \times 6080 \times 3}{60 \times 33,000 \times 2} = \frac{TS}{217}$$

$$T \text{ (Mean normal thrust)} = \frac{\text{I.H.P.} \times 217}{S}.$$

From the above it will be seen that the thrust varies directly as the indicated horse power and inversely as the speed. If the speed is decreased without a corresponding decrease in power, the thrust must increase. The actual thrust exceeds the normal when the engine first moves and the power is largely employed in overcoming the inertia of the ship. Also, when towing or driving against a head sea, so that the thrust of the same screw may vary considerably. The area of the collars of the thrust bearing are, however, calculated from the mean normal thrust and provision is made for all emergencies by making the total area of the thrust faces such that the pressure upon each square inch does not exceed seventy pounds.

If D is the diameter of the thrust collars,

d is the diameter of the shaft,

X is the number of thrust collars,

$$\text{then, } T = 70 \left(\frac{\pi D^2}{4} - \frac{\pi d^2}{4} \right) X.$$

Twin Screws.—Twin screws are exclusively used in all modern vessels in the naval service with reciprocating engines.

The advantages obtained from twin screw propulsion are as follows:

1. The duplication of machinery prevents a total collapse in case of accident; for with a break-down in one set of engines the other set could propel the vessel at a fair speed.

2. The propulsive efficiency is equal if not better than a single screw propulsion.

3. The maneuvering power of the ship is greatly increased.

4. In case of accident to the rudder the ship can be steered by her propellers alone.

5. Renders possible the use of vertical engines by reducing the height necessary. Two sets of engines running at a higher speed of revolution than a single engine developing the same power, enable a shorter stroke to be used and consequently a reduced height.

6. With vertical engines and twin screws the subdivision of the hull into longitudinal water-tight compartments is made practicable, so that if an engine room is flooded the other engine room could still be used, though the list might be serious.

7. Smaller diameters of screws can be used, hence, deeper immersion.

8. The smaller diameter of screws and their location with reference to the stern, tend to diminish the augmentation of resistance.

Triple Screws.—In this system three independent engines are fitted in three separate water-tight compartments and connected to three separate screws. The engines of the wing screws are arranged similarly to twin-screw engines, and abaft these on the center line are placed the engines for the center screw.

The question of twin screws vs. triple screws has become of little importance since the successful advent of turbines for marine propulsion, with which as many as four propellers on two or four shafts are often used.

Forms of Propellers.—Screw propellers have been made in the greatest variety according to the number, style, shape and arrangement of the blades.

The form of the blade is usually varied according to the view of the designer. It has not been found that the shape has a very great effect upon the propulsive efficiency, although it has some little influence upon the amount of vibration. The principal points to be considered in the design are the pitch and surface in relation to the diameter.

In modern practice the general shape of the blade is elliptical or oval, the broadest part being usually about half the extreme

radius of the screw, with a considerable variation both in outline and proportion as appears consistent with efficiency. Usually, the blades stand at right angles to the axis, though propellers are frequently found with the blades bent backward along a straight line or a curve, with a view to avoid the centrifugal action which conduces to an increase in thrust deduction. The pitch may be uniform or variable, the blades should have well-rounded ends with a large radius, in order to reduce the friction. On account of the increased angle of the blade as it approaches the axis, the inner part of the blade has but little propulsive effect and absorbs power in churning the water; for this reason the hub is enlarged and of a form to pass through the water easily.

Small propellers are usually cast complete in one casting. Large propellers are made with separate blades secured to the central hub by stud bolts and made capable of adjustment of the pitch within small limits. The advantages of the separate blades lie in the possibility of adjusting the pitch to suit the conditions and in the readiness with which repairs can be made. A broken or defective blade can be readily removed and replaced by a new one; in small vessels this can sometimes be done without going into dock. In all naval vessels extra blades are carried as spare parts, and can be connected up much more easily than an entirely new propeller.

The Griffith Screw.—In this screw the central inefficient portion is replaced by a large spherical hub about one-fourth to one-fifth of the diameter of the propeller. The outer ends of the blades moving with the highest velocity are narrowed to reduce the loss from friction, the outline of the blades being pear-shaped. The pitch near the boss is frequently made finer so that the angle at this part shall not be so coarse as only to churn the water.

In the navy the type of screw universally employed is the modified Griffith, and is one having blades of any curved shape diminishing in width towards the tips, having the broadest part nearer the boss than the tip, and of the general type illustrated in Plate 30, showing the screw employed in U. S. naval vessels.

For the method of designing and laying down a screw propeller in detail for a given ship, the student is referred to "Screw Propellers, Method of Design," by Captain C. W. Dyson, U. S. N. (See

book of plates.) The various methods of obtaining the dimensions of different parts as determined by different authorities are there given in detail, and their value compared with the screws placed on modern vessels in the navy. The following is a general scheme for the various parts; no exact rule can be laid down for guidance, as much must be left to the designer to suit the circumstances in each case.

Number and Pitch of Blades.—Two, three and four blades may be employed. In the navy three-bladed screws are in general use. As a rule the pitch ratio should be from one to 1.5 times the diameter. For modern vessels in the U. S. Navy the pitch averages about 1.05 times the diameter. A uniform pitch is desirable, as the advantages accruing from an expanding pitch are somewhat doubtful. In turbine screws the pitch ratio is .8 to 1.0.

Let S = The required speed of ship in knots per hour,

R = The designed number of revolutions per minute,

X = Slip per cent. In design a standard slip of 20 per cent allowed,

$$\text{Then, } S = \text{speed of screw} \times \left(1 - \frac{X}{100}\right),$$

$$\text{Speed of screw} = \frac{S \times 100}{100 - X},$$

$$\text{Pitch} = \frac{S \times 100 \times 6080}{60 R (100 - X)} = \frac{S}{R} \times \frac{10133}{100 - X}.$$

Diameter.—This must be less than the draught to prevent racing and grounding. The propeller must have sufficient immersion, for if it breaks the surface of the water, the efficiency is reduced to a remarkable extent. In practice for large vessels there should be four feet from the surface of the water to the blades and from $1\frac{1}{2}$ to 2 feet from the bottom of the keel to the bottom of the blade. Also, the diameter varies as the square root of the indicated horse power and inversely as the cube of the speed.

Material of Blades.—In the mercantile marine on account of cheapness, the screws are frequently made of cast iron and cast solid with the hub in one casting. In the navy, the propellers are made

of manganese bronze or any equivalent metal, such as phosphor bronze, monel metal, etc.

By the use of alloys, such as manganese bronze, the blades can be made much lighter than those of cast iron, and the edgewise resistance is thus reduced. In some cases cast steel blades have been used, in order to reduce the thickness, but with this material excessive corrosion has occurred on the backs of the blades due to the air detached from the water.

In propellers with separate blades, the root of the blade terminates in a circular flange, fitting into a circular recess in the hub, as shown in Plate 30. The blade is secured to the hub by means of Tobin bronze stud bolts *A*, passing through the flange, the latter being recessed so that the securing stud bolts come below the surface of the hub and are covered with composition plates or covers *B*, held by countersunk screws and finished to form a smooth surface fair with the boss. By this means the spherical form of the hub is maintained and friction through the water reduced.

The bolt holes in the flanges of the propeller blades are made oblong to allow the adjustment of the pitch, by turning the propeller blade slightly in one direction or the other, and thus changing the obliquity of the blade to the axis of the propeller shaft. After the pitch is set, packing pieces of brass or lignum vitæ are filled in on each side of the bolt to prevent any change from the blade slipping from its position. The securing bolts are prevented from turning by means of lock plates *C* fitted in the recesses for the bolt heads as shown in Plate 30. In shifting the blade for an alteration of pitch it should be noted that the change of pitch resulting from the angular change is not the same for all parts of the blade, but varies from root to tip. For the moderate alterations made, however, the departure from uniformity is inconsiderable.

The acting surface *D* or front of the blade only preserves its exact helical form, the back *E* is modified to give the required thickness at the different parts and is not a true screw surface. The transverse sections *FFF* approximate semi-ellipses, shallow at the ends and increasing towards the root, as shown in Plate 30.

The boss is accurately fitted to the taper on the after end of the propeller shaft and to a longitudinal key *G* in the shaft with a cor-

responding recess in the boss. The shaft casing enters into the boss about one inch, and is fitted perfectly water-tight. The propeller is held on the shaft by a nut screwed on and locked in place and the nut is covered by a composition cap *H*, bolted water-tight to the boss to prevent the corrosive action of the water on the end of the shaft. In order to reduce the liability of the securing nut loosening or backing off, for a right-hand propeller the thread of the

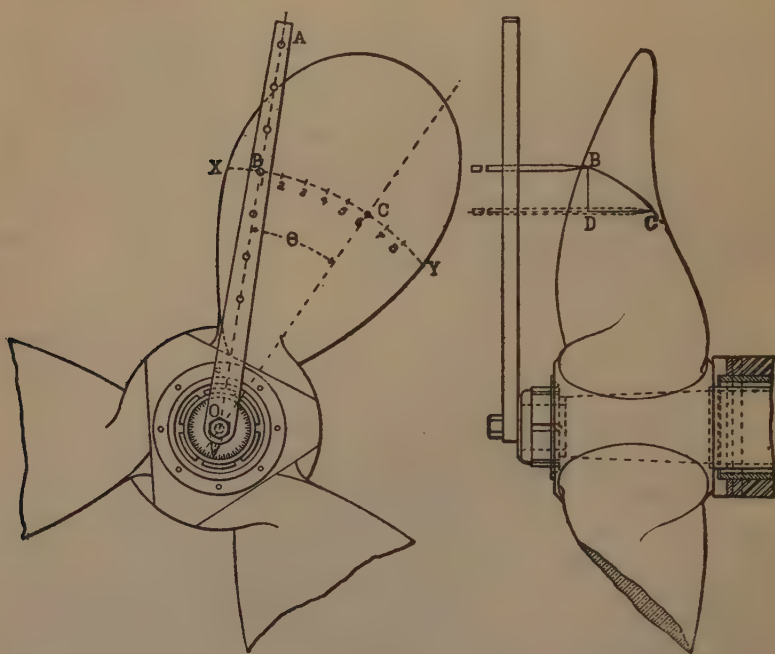


FIG. 174.

nut is left handed, and vice versa. The cap is conical in shape to reduce the eddying loss, and as a preventive of fouling ropes, etc.

The boss and cap are finished all over and the blades smoothed by burnishing. After the propeller is connected up it is usually swung on a mandrel or between centers, and balanced by removing excess of metal on heavier blades.

Practical Methods of Pitch Measurement.—In estimating the pitch of a propeller three measurements are required:

1. The radius at which the pitch is to be measured.
2. The angle between the two points between which the pitch is required.
3. The axial advance corresponding to the angle or part of the circumference moved through between the two points taken above.

These measurements are taken as follows: Remove the cap over the nut at the end of the shaft and secure a batten OA at a point corresponding to the center of the shaft and to revolve in a plane at right angles to the axis of the shaft. In Fig. 174 the batten has holes drilled along its length at equidistant points from the axis, through which can be passed a rod graduated in inches and which can be clamped in any desired position. This rod must always be parallel to the axis of the shaft, so that the arcs described in rotating it about the center shall always correspond to the parts of the circumference due to the radial holes on the batten. By means of the rod and batten, describe on the face of the blade an arc such as XY ; denote the radius taken by r . Then on this arc described, take any two points, such as B and C , between which the pitch is to be measured. By means of the rod measure the distance between the batten and face of the blade at the point B . Swing the batten around to the point C , noting the angle θ made between the two points and measure the vertical distance between the batten and the face of the plate at C . The difference between these two measurements ($C - B$) is the axial distance made good by the point B in revolving through an angular distance θ . Since in one complete revolution the point B travels a distance equal to the whole pitch and for an angular movement θ , a measured distance equal to $C - B$, we have,

2π or 360° : whole pitch (P) :: angular movement between B and C (θ) : fraction of pitch ($C - B$).

$$\therefore \text{Pitch} = \frac{2\pi}{\theta} \times (C - B).$$

For example, by measurement of axial distance between two points on the blade at a given radius has been found to be $(17\frac{1}{2}'' - 3\frac{1}{8}'') = 14\frac{3}{8}''$ through an angle of 30° .

Then pitch = $14\frac{3}{8} \times \frac{360}{30} = 172\frac{1}{2}'' = 14\frac{3}{8}$ feet.

When $\theta = 30^\circ$, $C - B$ measured in inches gives the pitch of the screw at once in feet.

The operation is precisely the same for a uniform or a variable pitch, and measurements should be taken at different parts of the same radius and at different radii, to ascertain whether or not the pitch is variable, either axially or from leading to following edges. The mean of the pitches thus obtained gives the approximate mean pitch of the blade.

If it is inconvenient to measure the angular movement of the batten, the length of the arc subtending θ in a plane perpendicular to the axis may be obtained as follows:

In Fig. 174, BC is the arc on the blade with radius r , and DC is the corresponding fraction of the pitch ($C - B$). BD is the arc subtending θ , the angular distance moved by the batten. Measure the arc BC on the blade and the fraction of pitch used ($C - B$), then the arc BD subtending $\theta = \sqrt{BC^2 - CD^2}$ and pitch =

$$\frac{2\pi r \times CD}{\sqrt{BC^2 - CD^2}}.$$

For a rough method of quickly estimating the pitch of a true screw, take that part of the blade where the inclination of the face is 45 degrees to the axis of the shaft, then the pitch will be equal to the circumference due to that radial distance.

Details of the above principle for finding the pitch may be varied to a large extent to suit the conditions of the case, and generally the required measurements can readily be obtained.

Another application, as given by Barnaby, is as follows: Divide the arc XY (Fig. 174) into a number of equal parts, 1, 2, 3, 4. Measure off the same number of intervals upon a base line $X'Y'$ equal in length to the developed arcs, Fig. 175. Measure the ordinates at such intervals on the arc from the batten or a plane perpendicular to the axis and lay off the ordinates 1-1', 2-2', 3-3', etc., at the corresponding intervals on the base line. Through the points 1', 2', 3', etc., draw the line AB produced to cut the base line at A . With a uniform pitch, AB will be a straight line. From A lay off AC equal to the circumference of a circle at the radius from which measurements are taken, then the perpendicular DC

on AC will give the pitch of the screw. With a variable pitch, the points $1'$, $2'$, $3'$ will lie in a curve (Figs. 176 and 177). To find the mean pitch in this case, draw a tangent to the extremities of the curve and take the arithmetical mean of the two pitches obtained.

In-Turning and Out-Turning Propellers.—The question as to which direction it is desirable to make twin screws turn came under discussion soon after twin screws were installed, and different phases of the question have been brought up from time to time. For many years the propellers were so arranged that the upper tips of the blades moved outward or away from the hull when going ahead. Experimental work at the Admiralty Experimental Tank, under

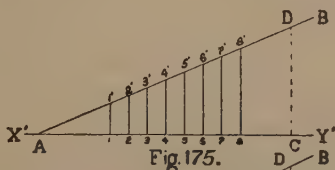


Fig. 175.

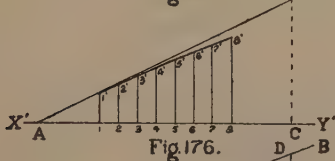


Fig. 176.

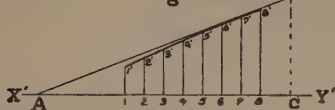


FIG. 177.

the direction of Mr. R. E. Froude, in 1898, indicated a slight increase in efficiency by having the screws turned in the opposite direction or towards the hull in going ahead, so that in the British Navy new ships were largely fitted with in-turning screws. Other experimental work seemed to favor the in-turning screw for efficiency, but this data is not conclusive; it is believed that the out-turning screw is practically as efficient, and this type is exclusively fitted now in our navy.

With the propellers to turn towards the hull the engines can be conveniently installed in such a manner as to locate the working platforms in front of the engines and near the center lines. This

is an advantageous arrangement for ease of operation and in the interchange of orders between the engine rooms, besides insuring greater convenience of installation and better conditions of ventilation for the working platforms.

The installation of in-turning screws has brought out much adverse criticism as to the maneuvering qualities of ships so fitted, particularly with the destroyer and torpedo-boat type, where the shafts are close together, the tips of the blades when in-board coming extremely near each other, and the blades when upward being close to the surface of the water, and it is regarded as conclusive in our navy that out-turning screws give better maneuvering qualities. Advocates of the in-turning screw claim that with deep-draught battleships and cruisers, the shafts well separated, propellers deeply immersed and the action of one propeller uninterfered with by the other, the direction of the turning of propellers has but little, if any, influence upon the maneuvering qualities.

Standardized Propeller Speed Trials.—Under this system a vessel must first make a progressive speed trial for standardization. For this purpose the vessel makes five runs consecutively in alternately opposite directions at highest speed over a measured course, preferably one mile, the speed and revolutions being accurately recorded. Additional points are obtained in similar manner in order to construct a curve—except that only three alternating runs are made at each speed, instead of five; at least three other points are obtained at near top speed, say about $\frac{1}{2}$ knot, 2 knots, and 3 knots less than full speed, and three or more others at intervals down to about 8 knots. Data for a speed-revolution curve being thus obtained, the vessel is run continuously in deep water for the time required by the contract, and the speed is then calculated from the revolutions made.

From indicator cards taken during these runs the I.H.P. is found and a speed-I.H.P. curve also plotted.

Formerly the "preliminary acceptance" trials to determine the speed of vessels before acceptance from the contractors were run over a measured course for four hours, during which time the average speed must be corrected by tidal corrections. These corrections necessitated extensive observations of current by station boats along the course, and foggy weather often interfered with trials. The standardized screw method is now used.

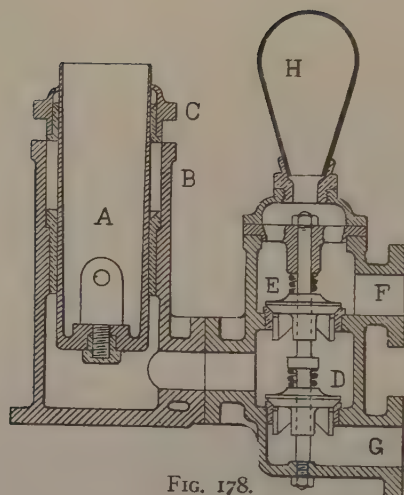


FIG. 178.

CHAPTER XIV.

AUXILIARY MACHINERY.

Pumps.—The feed water is forced into the boiler by pumps or by injectors. The former are divided into two general classes, *plunger pumps* or outside packed pumps, and *piston pumps* or inside packed water pistons.

The simplest type of the plunger pump, frequently used for a bilge pump, is driven by an eccentric on the main shaft, the eccentric in this case acting as a crank, and is shown in Fig. 178. The plunger is shown at *A*, the packing box at *B*, and packing gland at *C*. There are two valves or sets of valves, the lower ones *D*, are the foot or suction valves, and the upper set, the head, delivery or discharge valves, *E*. The action of the pump is as follows: As the plunger rises, producing a partial vacuum in the space underneath, the water flows into the pump barrel from the bilge, by way of *G* and the foot valves *D*, opening upward. On the return stroke the water contained in the pump barrel is compressed, the foot valves close, and the water is forced out through the discharge valves *E* and through the passage *F* to the discharge. The pump thus delivers but once in two strokes and is known as a *single-acting pump*. The suction channel *G* is usually placed in a bilge well in order that the water may flow easily and quickly to the pump barrel as the plunger rises. An air chamber *H* is attached to the discharge chamber of the pump, and acting as a cushion, prevents sudden shocks on the mechanism by the reversal of the flow of water. The water partially filling the chamber, compresses the contained air and forming an elastic cushion, takes up the shock as the body of water under heavy pressure is forced into the discharge pipe. An air chamber is often placed on the suction pipe of large pumps for the same purpose. The only place that leakage can occur is at the plunger packing, which is accessible from the outside. The suction and delivery

valves and their seats can be examined and overhauled by the removal of bonnets from the valve chest.

The stuffing box is packed with rings of hemp, flax, or rings of canvas rolled around a central core and set up by the gland and nuts.

Direct-Acting or Positive Motion Pumps.—For many years during the era of moderate pressures and speeds, pumps for feed and general purposes were employed fitted with a crank shaft together with an eccentric for regulating the movement of the steam valve, and a fly wheel for carrying the pump over the dead points. As high

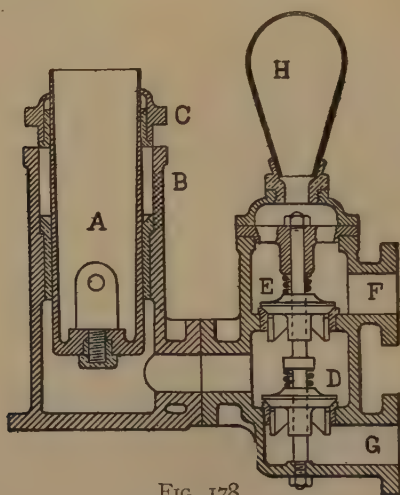


FIG. 178.

steam pressure came into use, this pump was replaced with a direct-acting pump of larger size, slower speed, greater compactness in form, and better adapted to the service required.

Owing to the absence of rotating parts in these pumps, many ingenious devices have been employed to regulate the steam valve movement, and giving rise to the many excellent pumps in commercial use. Those usually employed in the U. S. Navy are the Blake, Davidson, Worthington, Blake-Knowles and others. These pumps of the different makers named can be seen either in section or in operation in the Model Room and Laboratory of the Engineering Building.

Since an eccentric cannot be employed, the usual arrangement in the interior of the steam chest consists generally of a main valve, a supplemental valve regulating the steam and exhaust movements

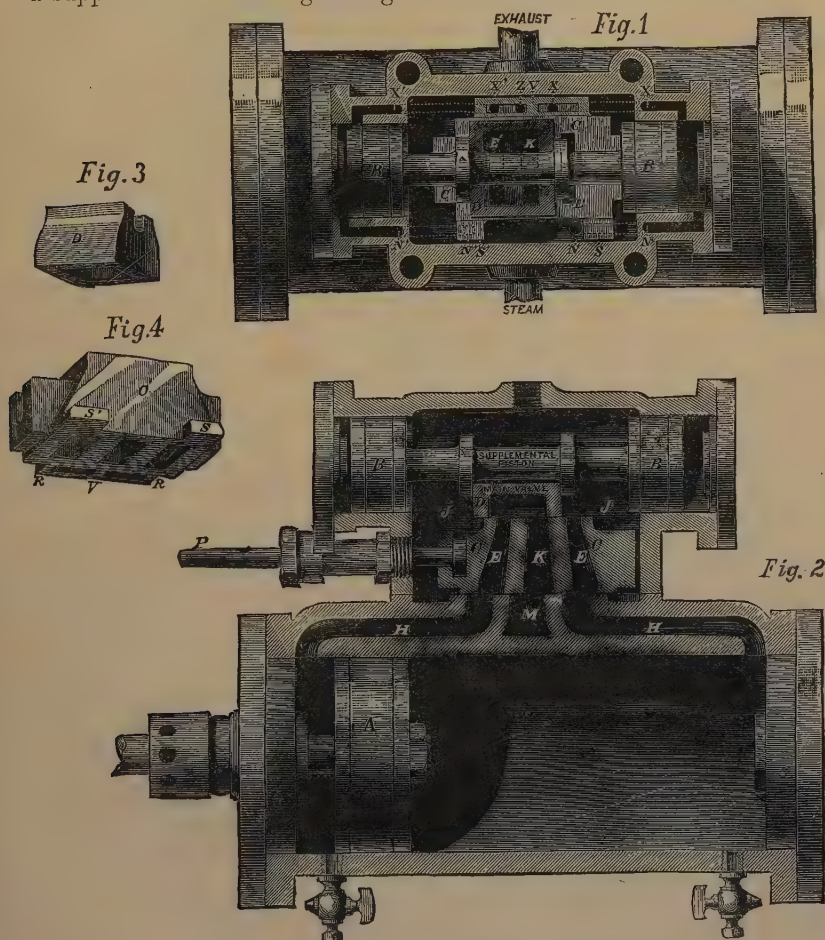


FIG. 179.

of a piston in a cylindrical cavity in the valve chest, and connected to and controlling the movements of the main valve. Motion is given the supplemental valve by means of links, rods, tappets, etc., on the piston rod. The position of the tappets may be moved so as to give a longer or shorter main piston stroke.

A description in detail of the action of the valve gear of one type (the Blake) will be given below, while an analysis of the others can be made in the course of Practical Drills in the Engineering Department.

Referring to Fig. 179, showing sectional views of the steam cylinders, valves, etc., *A* is the main piston, *B* the supplemental or valve driving piston, *D* the main valve divided into two parts, the lower part *C* sliding upon the valve seat, and with ports passing directly through the valve, while *D* slides upon the upper face of *C*.

As shown in the cut, *A* is approaching the left-hand end of its stroke, and *C* at the opposite end of its stroke, steam from the steam chest *J* entering the right-hand end of the main cylinder through the ports *E* and *H*, and the exhaust escaping through *H'*, *E'*, *K* and *M*. When the main piston *A* has nearly reached the left-hand end of its stroke, a connection of the piston rod (see Fig. 183) strikes a tappet on the valve stem *P* and moves the latter, together with *C* and its supplemental valves *R* and *SS'* (which form with *C* one casting) to be moved from right to left. This movement of *C* admits steam to the left-hand end of the supplemental cylinder and the piston *B* will be forced to the right, carrying the main valve *D* with it to the right-hand end of its stroke; for the movement of *S* closes *N* (the steam port leading to the right-hand end), and the movement of *S'* opens *N'* (the steam port of the left-hand end) at the same time the movement of *V* opens the right-hand end of this cylinder to exhaust through the exhaust ports *X* and *Z*.

All parts of the valve and connections now have opposite positions to that shown in Fig. 179, 1 and 2, and steam is therefore entering the main cylinder through *E'* and *H'* and exhausting through *H*, *E*, *K* and *M*, thus causing the main piston to move in the opposite direction, or from left to right. Similar operations to those already described will follow when the piston approaches the right-hand end of the cylinder. By this arrangement the pump is rendered positive in its action, that is, it will instantly start in any position and continue working the moment steam is admitted to the steam chest.

The valve gear above described will be found on many pumps in use, but the most modern type now used on the Blake-Knowles

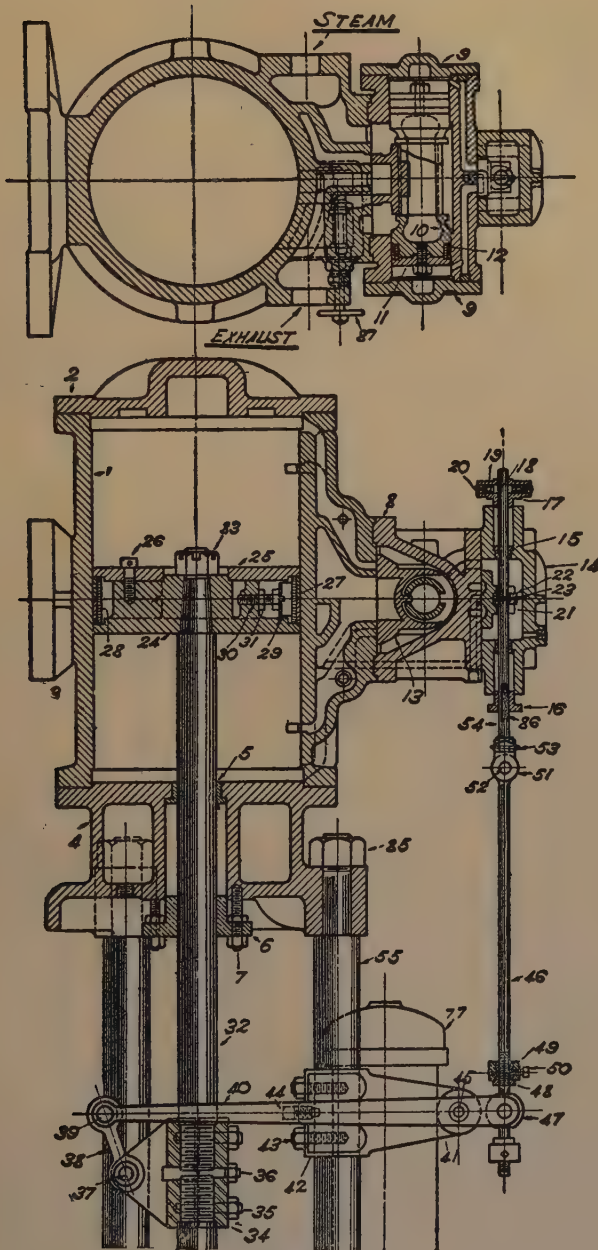


FIG. 180.

pumps is shown in Fig. 180. Here the auxiliary valve 21 works on a seat outside of the auxiliary plunger 10 and in a separate valve chest. The seat upon which it works has four ports, two to each end of the auxiliary plunger, one being a live steam port and the other the exhaust port. The cushioning of the auxiliary plunger is accomplished by the plunger cutting off its own exhaust port which enters the auxiliary cylinder some distance back from the head, as shown in the figure. After cutting off this port it cushions on the steam confined in the cylinder between the exhaust port and the head and also the steam in the live steam port. The live steam which operates the plunger enters through two ports in the very extreme ends of the auxiliary cylinder; these ports are not shown in the cut.

The valve gear is shown in mid-position.

By means of the small auxiliary valve, receiving its motion from the piston rod, the auxiliary plunger is supplied with steam, and this in turn moves the main slide valve back and forth, distributing steam to the pump.

The new valve gear for Davidson steam pumps is shown in Fig. 181. The cam and pin heretofore used to actuate the valve (shown in Fig. 182) is lacking, the movement of the latter being obtained by alternately connecting the ends of the valve chest to the exhaust. The auxiliary valve *A* is drilled out axially and radially, as shown, to afford ready connection to the exhaust. The pump piston is shown nearing the top end of its stroke; upon rising a little higher the exhaust is connected through the auxiliary valve passages to V^1 —and the main valve piston, now at the right end of the stroke, admitting steam to the bottom of pump piston and exhausting from the top, is driven over to the left, causing the pump to start its down stroke.

The water ends of the different pumps are very similar. In Fig. 182, showing a sectional view of a Davidson pump, *G* is a water piston driven by a prolongation of the steam piston rod. While the water piston moves to the right a partial vacuum is formed behind the piston and water is drawn from the source of supply through the passage *H*, the suction valve *V* and passage *K* to the pump barrel. At the end of the stroke the motion is reversed and

the piston starts to return, compressing the water behind it and forcing it up through the port *K*, on top of and closing the valve *V*, and opening the discharge valve *L*, through which the water enters the discharge chamber and out into the feed line. This operation is repeated for each end of the stroke and as there is a delivery of

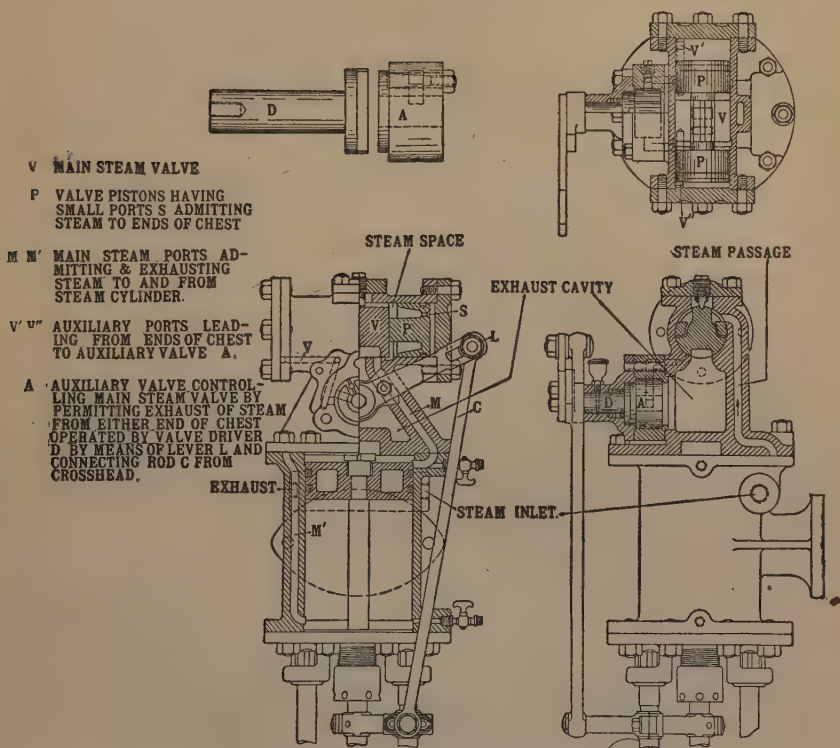


FIG. 181.

water for each stroke, the pump is known as a *double-acting pump*. The total pressure on the steam piston is equal to the steam pressure multiplied by the area of the piston in inches, while that on the water piston is equal to the water pressure multiplied by the area of the water piston, hence, by varying the ratio of the areas of these two pistons, the water pressure can be made so much higher per

square inch that it will not only overcome the friction of pipes, valves and passages, but will force the discharge water into the boiler against the steam pressure which drives the pump, or into a boiler with a pressure even higher than that from which it is taken. For feed pumps the area of the steam piston is made from two to three times the area of the water piston.

With feed and fire service pumps, in order to provide for a relief from an excessive pressure in the piping, in case of failure to open check or discharge valves when the pump is in operation, it is cus-

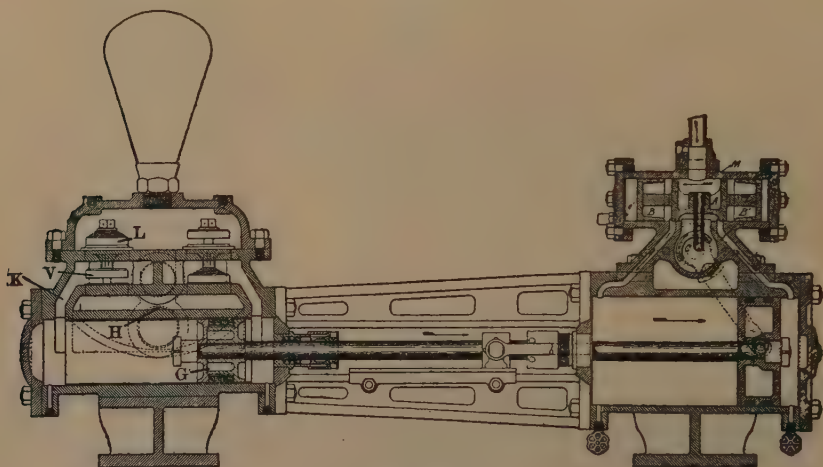


FIG. 182.

tomary to fit a safety relief valve to the discharge chamber of the pump, or the discharge pipe, and this usually consists of a simple form of spring-loaded safety valve.

Duplex Pumps.—Frequently two pumps are placed side by side and the valve movement of one pump is actuated by connections with the piston rod of the other pump, as shown in Fig. 183. This form of pump is known as a *duplex* pump, in contradistinction to the pump previously mentioned called a *single* or *simple* pump.

For naval use, on account of the less floor space required and the more convenient arrangement for quick examination of the valves, the vertical style of pump is now universally used.

For feed purposes, either plunger or piston pumps are now specified for all recent installations in the ships under construction

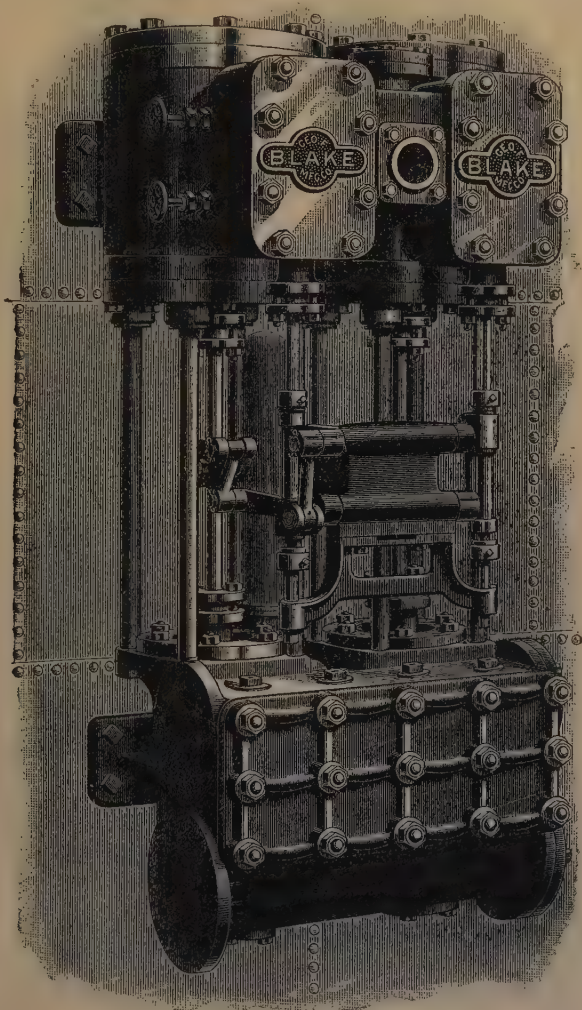


FIG. 183.

for the U. S. Navy. A typical example may be seen in Figs. 184 and 185 and consists of a vertical plunger, double acting, single pump, outside center packed.

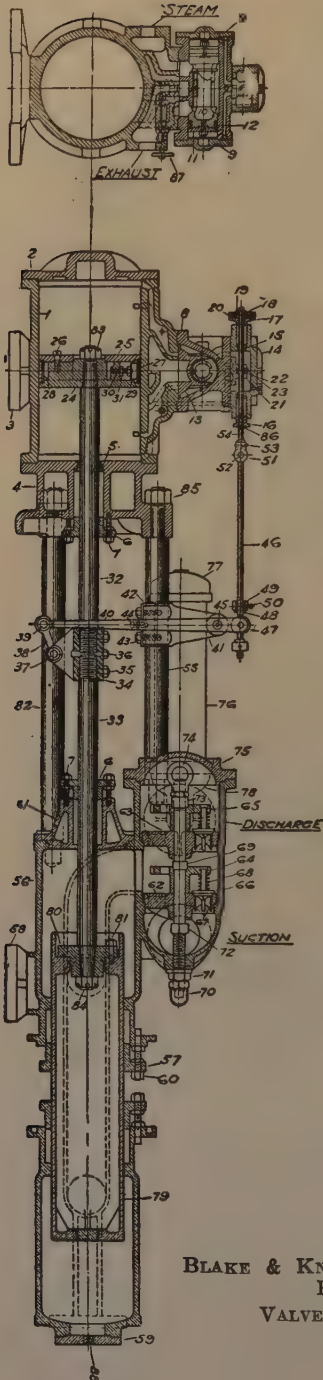


FIG. 184.

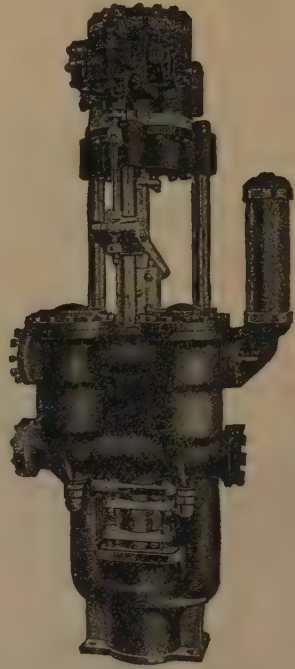


FIG. 185.

BLAKE & KNOWLES VERTICAL SIMPLEX CENTER
PACKED PLUNGER PUMP.
VALVE POT WATER CYLINDER TYPE.

No.	Name of Part.	No.	Name of Part.
1	Steam cylinder.	46	Lower valve rod link.
2	Upper steam cylinder head.	47	Tappet.
3	Steam cylinder foot.	48	Valve rod collar split bushing.
4	Lower steam cylinder head.	49	Valve rod collar.
5	Throat bushing.	50	Valve rod collar set screw.
6	Piston rod gland.	51	Valve rod knuckle.
7	Piston rod gland stud and nut.	52	Valve rod knuckle pin.
8	Auxiliary steam cylinder.	53	Valve rod knuckle taper pin.
9	Auxiliary steam cylinder heads.	54	Valve rod.
10	Auxiliary steam cylinder plunger.	55	Front tie rod.
11	Auxiliary plunger follower.	56	Plunger barrel.
12	Auxiliary plunger rings.	57	Plunger gland.
13	Main slide valve.	58	Pump cylinder foot.
14	Auxiliary steam chest.	59	Lower pump cylinder head.
15	Valve rod box throat bushing.	60	Plunger gland stud.
16	Valve rod gland.	61	Upper pump cylinder head.
17	Valve rod friction gland.	62	Suction valve plate.
18	Friction gland check piece.	63	Discharge valve plate.
19	Friction gland spring.	64	Suction valve spring guard.
20	Friction gland set screw.	65	Discharge valve spring guard.
21	Aux. steam chest slide valve.	66	Valve.
22	Valve rod block.	67	Valve seat.
23	Valve rod block taper pin.	68	Valve spring.
24	Steam piston head.	69	Valve plate rod.
25	Steam piston follower.	70	Valve plate cap nut.
26	Steam piston follower bolt.	71	Valve plate flanged nut.
27	Steam piston ring, outer.	72	Valve plate nut.
28	Steam piston ring, inner.	73	Valve plate guard nut.
29	Steam piston spring.	74	Valve plate rod eye.
30	Steam piston set-out bolt.	75	Valve chamber cap.
31	Steam piston set-out check nut.	76	Discharge air chamber.
32	Steam piston rod.	77	Discharge air chamber cap.
33	Pump piston rod.	78	Discharge air chamber elbow.
34	Crosshead.	79	Plunger.
35	Crosshead studs and nuts.	80	Plunger head.
36	Crosshead taper pin.	81	Plunger studs and nuts.
37	Crosshead pin.	82	Side tie rod.
38	Lower link.	83	Steam piston rod nut.
39	Lever pin.	84	Pump piston rod nut.
40	Lever.	85	Tie rod nut.
41	Pedestal.	86	Valve rod gland stud and nut.
42	Pedestal cap.	87	Cushion valve complete for steam cylinder.
43	Pedestal studs and nuts.	88	Plunger barrel drain plug.
44	Pedestal set screw.		
45	Pedestal pin.		

The sectional view is taken through the central line of the steam and water cylinders, and through one suction and delivery valve chamber, showing the entrance and exit of water drawn to and discharged from the upper part of the pump barrel. Similar sets of valves are arranged in a chamber just behind this, providing for the entrance and delivery of water for the lower part of the barrel. The principle is shown in Fig. 186.

The water enters the common passage under the suction valves for both ends through the suction opening *A*. As the plunger rises

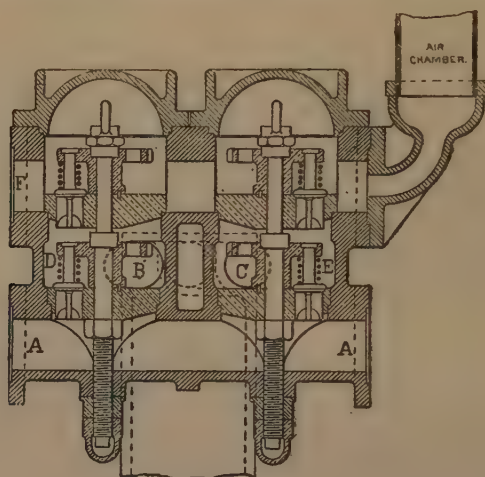


FIG. 186.

the suction valves open and water passes into the lower part of the pump barrel, through the passageway *B*. At the same time the water in the upper part of the barrel, above the plunger, is being forced through the passage or port *C*, on top of the suction valves in the opposite chamber *E*, closing them and opening the delivery valves of that chamber, and forcing the water out through the common delivery pipe of both ends. On the down stroke this operation is reversed, the upper end takes the suction through its suction valves and the port *C*, the plunger forces the water supplied to the lower part of the barrel into the passageway *B* over the suction valves, under the delivery valves, forcing them open and into the common discharge pipe.

For independent bilge pumps and for all general or light service pumps in the navy, the direct-action piston pumps of the general type shown in the preceding are employed and made either single or duplex. For pumps used on the fire main where a considerable pressure is required, the ratio of piston areas as used for the feed pumps is usually employed. For light service, such as pumping the bilge, distiller circulating pumps, flushing, etc., the area of the steam piston is but little in excess of that of the water piston.

The commercial method of designating pumps is by diameter of steam piston, diameter of water piston, and the stroke. Thus a catalogued pump 8" x 5" x 10" is a pump whose steam piston is 8 inches in diameter, water cylinder 5 inches in diameter, and 10 inches stroke.

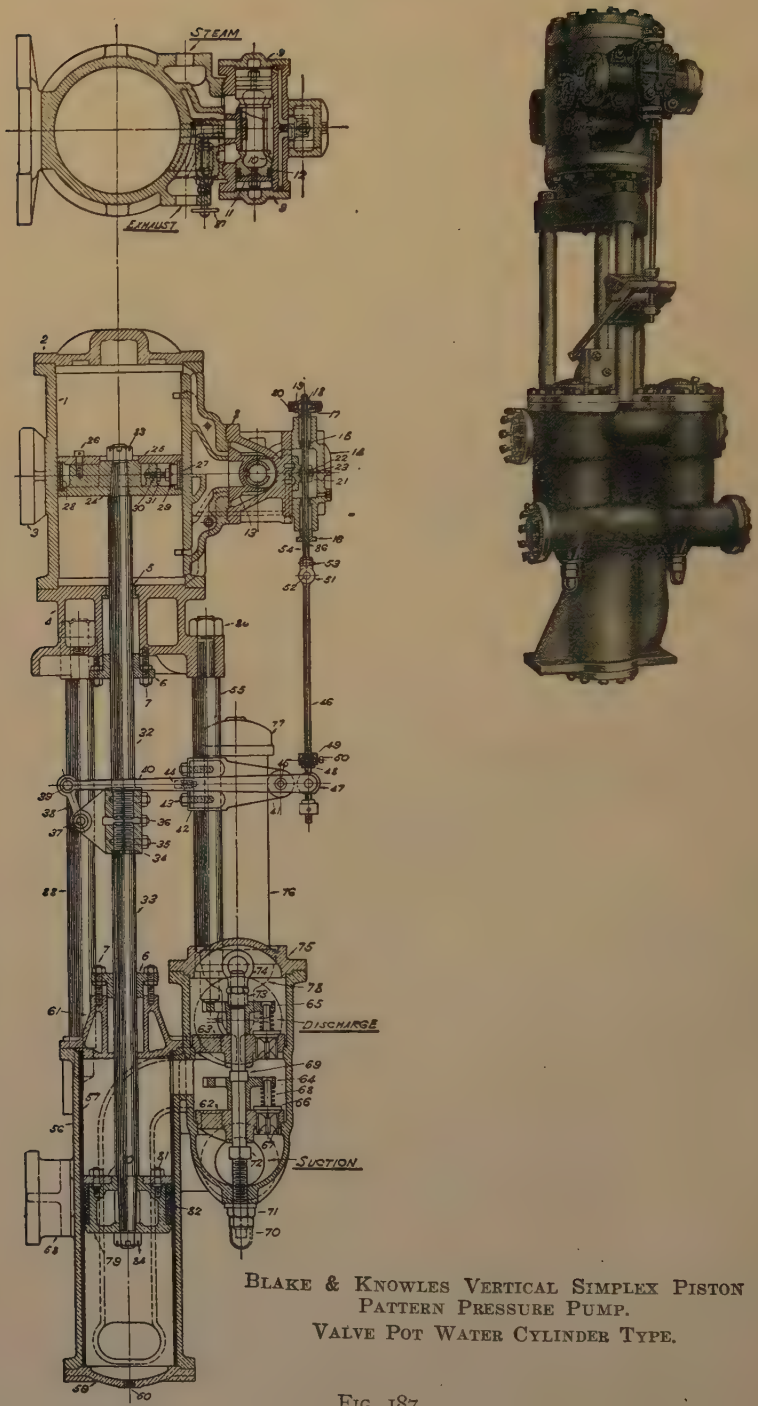
The piston type of pump, also used for feed purposes, is illustrated in Fig. 187.

The same size pumps are usually employed for the auxiliary feed pumps as for the main feed. The main feed pumps have but one suction, viz., to the feed tanks, and a discharge to the feed system, and are employed for boiler feed purposes only.

The suction usually connects with main condenser channel-way (air pump suction). There are usually four main feed pumps in large ships, located in the engine rooms.

One auxiliary feed pump is located in each fireroom, having suctions to main feed tanks and condenser channel-way, to reserve feed tanks, to C. & R. pipes leading from ship's side (one pump only, to take fresh water), and to hose connection. The auxiliary feed pumps deliver to main feed line, to hose connection to reserve feed tank (only the one having suction to ship's side), and to each boiler, in compartment where pump is located, through auxiliary feed check valve.

Centrifugal or Turbine Feed Pumps.—Reciprocating piston or plunger pumps are still solely used in our Navy for boiler feed pumps, but the centrifugal pump is much used in stationary plants in recent years. Its ability to deliver water against any desired pressure is acknowledged; by simply increasing the number of stages the delivery pressure is increased. In some land pumping plants as many as twelve stages are used, the delivery pressure reaching to



BLAKE & KNOWLES VERTICAL SIMPLEX PISTON
PATTERN PRESSURE PUMP.
VALVE POT WATER CYLINDER TYPE.

FIG. 187.

No.	Name of Part.	No.	Name of Part.
1	Steam cylinder.	46	Valve rod link.
2	Upper steam cylinder head.	47	Valve rod tappet.
3	Upper steam cylinder foot.	48	Valve rod collar split nut.
4	Lower steam cylinder head.	49	Valve rod collar.
5	Throat bushing.	50	Valve rod collar set screw.
6	Piston rod gland.	51	Valve rod knuckle.
7	Piston rod gland stud.	52	Valve rod knuckle pin complete.
8	Auxiliary cylinder.	53	Valve rod knuckle taper pin.
9	Auxiliary cylinder head.	54	Valve rod.
10	Auxiliary cylinder plunger.	55	Front tie rod.
11	Aux. cyl. plunger follower.	56	Pump cylinder.
12	Auxiliary cylinder plunger rings.	57	Pump cylinder lining.
13	Main slide valve.	58	Pump cylinder foot.
14	Auxiliary steam chest.	59	Lower pump cylinder head.
15	Valve rod box throat bushing.	60	Lower pump cyl. drain plug.
16	Valve rod gland.	61	Upper pump cylinder head.
17	Valve rod friction gland.	62	Suction valve plate.
18	Friction gland check piece.	63	Discharge valve plate.
19	Friction gland spring.	64	Suction valve spring guard.
20	Friction gland set screw.	65	Discharge valve spring guard.
21	Auxiliary steam chest slide valve.	66	Pump valve.
22	Valve rod nut.	67	Pump valve seat.
23	Valve rod nut taper pin.	68	Pump valve spring.
24	Steam piston.	69	Valve plate rod.
25	Steam piston follower.	70	Valve plate rod cap nut.
26	Steam piston follower bolt.	71	Valve plate rod flanged nut.
27	Steam piston ring, outer.	72	Valve plate rod nut.
28	Steam piston ring, inner.	73	Disch. valve spring guard nut.
29	Steam piston spring.	74	Valve plate rod eye.
30	Steam piston set-out bolt.	75	Pump valve chamber cap.
31	Steam piston set-out check nut.	76	Discharge air chamber.
32	Steam piston rod.	77	Discharge air chamber cap.
33	Pump piston rod.	78	Discharge air chamber elbow.
34	Crosshead.	79	Pump piston.
35	Crosshead studs and nuts.	80	Pump piston follower.
36	Crosshead taper pin.	81	Pump piston follower stud.
37	Crosshead pin complete.	82	Pump piston packing rings.
38	Lower link.	83	Steam piston rod nut.
39	Lower link pin complete.	84	Pump piston rod nut.
40	Lever.	85	Tie rod nut.
41	Valve gear pedestal.	86	Valve rod gland stud.
42	Valve gear pedestal cap.	87	Cushion valve complete for steam cylinder.
43	Pedestal studs and nuts.	88	Side tie rods.
44	Valve gear pedestal set screw.		
45	Fulcrum pin complete.		

a thousand pounds. The characteristics of this type are described in the text-book on "Boilers." Some of the advantages claimed are:

Steady delivery pressure, causing less shock and vibration to piping.

Practically constant delivery pressure, which can never increase beyond a pre-determined maximum, thus obviating all danger of leaky joints or other injury even should all check valves be closed without stopping the pump.

No danger from racing of pump due to loss of suction. However, running at speed with all delivery shut off will shortly generate unsafe temperature from internal friction of the churned water.

Simplicity of construction, small number of parts, with consequent less repairs needed.

It can be more quickly started and put into service from a cold condition than a reciprocating pump.

Either electric motor or turbine drive may be used. As to efficiency, experiments do not show any decided superiority of the centrifugal pump in this respect as compared with a reciprocating pump in good condition. Probably it will remain in efficient condition longer, however, and therefore under service conditions may show superior economy.

Injectors.—In addition to the feed pumps, injectors are often fitted as an auxiliary for feeding the boilers; they are compact and efficient, and of value for boiler feed purposes, since the heat of the steam used in operating them passes into the feed water.

By means of this instrument a jet of steam from the boiler mixes with itself a stream of cold water, by which it is condensed, and to which it imparts so much of its own velocity that the combined mass of cold water and condensed steam enters the boiler against its own original pressure. There are many different forms of injectors, and a description of one will illustrate the principle employed in all. Referring to the diagram, Fig. 188, steam from the boiler enters through the pipe *B* and is admitted to the nozzles *S* and *V*, the opening for the steam jet being regulated by the handle *K*. The shape of the nozzle is such as to permit of the gradual increase of the velocity, so that on passing through the

suction nozzle V , a vacuum is formed and water is drawn up through the suction pipe B^1 with a great velocity, and streams through the combining nozzle C , its velocity being increased by the impact of steam on the back of the column. In the delivery nozzle D the stream expands as the cross-section increases; it therefore loses velocity but gains in pressure, until at the end of the tube the pressure is so great that it lifts the check valve on the discharge pipe and the water enters the boiler. An orifice surrounds the end

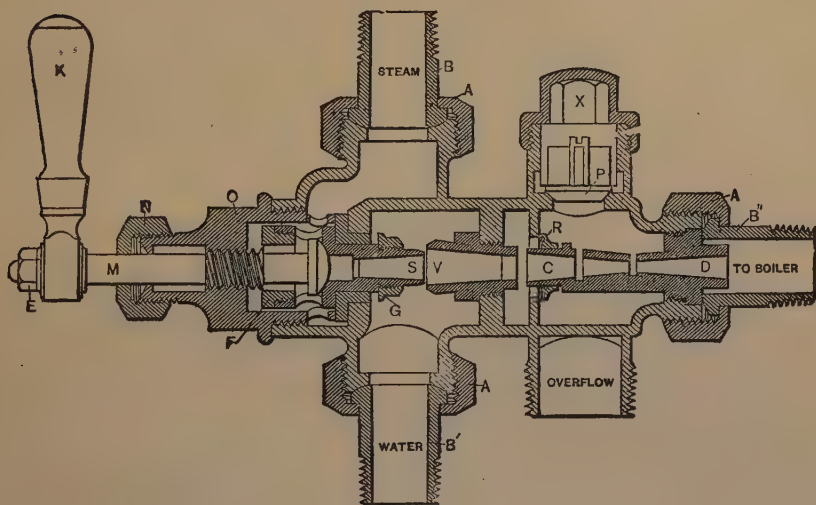


FIG. 188.

of the combining nozzle fitted with a valve *P* which opens communication with the overflow pipe. By this means the injector can start in operation by providing a channel through which steam and water may escape until the stream acquires sufficient energy to force its way into the boiler.

Steam expanding freely in the diverging nozzle *S*, when the injector is first started with the overflow open, acquires a high velocity, and consequently kinetic energy, and upon drawing in water from *B'* is condensed, forming a partial vacuum; the continuous formation of this partial vacuum affords the necessary

opportunity for expansion of the steam and consequent acquisition of high velocity and kinetic energy after the overflow is closed. It is easy to see that with equal pressure in the steam pipe *B* and in the boiler feed pipe *B''*, and with steam rushing through *S* to fill the vacuum caused by condensation in *V*, there must be a resultant of kinetic energy in the combined steam and water at *V* (since no kinetic energy is lost by condensation) directed toward the boiler check-valve—in other words, pressure. This is the case, and feed water can be forced into a boiler carrying much higher pressure even than that of the steam supply to the ejector.

Steam cutters are sometimes supplied with an *ejector*, shown in Fig. 189, for freeing the bilge of water. Its action is similar to that of the injector. Steam enters through *B* and a vacuum being



FIG. 189.

created by the steam rushing through the nozzle, the water in the bilge is drawn in through *A* and forcibly expelled overboard through the pipe connection with *E*. The objection to this apparatus is the loss of fresh water in operating the ejector. The hydrokineter for marine boilers is another application of the principles of the injector.

Feed Water Heaters.—The feed water heater has lately taken a very prominent part among the auxiliaries of a steamship, and its importance is due not only to the economy derived from its use, but also to its beneficial effects towards boiler preservation. Even when live steam is taken from the boiler direct, an appreciable gain in economy is obtained.

Feed water heaters are of two general types, according as the heat employed is drawn from the waste flue gases or from steam. In the first system a series of tubes are placed in the up-take above

the boiler, through which the feed water circulates, while the waste gases surround the tubes on their way to the chimney. Difficulty is experienced in this type; the efficiency of the heater becomes much impaired from the deposit of soot, it is not very accessible for repairs, and is not employed in the naval service.

In the second type of feed water heating, the heat given to the water is taken from steam drawn directly from a steam pipe, from one of the receivers, or from the exhaust from the auxiliaries. There are two methods in use on this principle. In one, the *open type*, the steam and feed water are mixed together in the same chamber, the steam condensing, gives up its heat to the feed and joins with the feed for a direct line to the boiler. In the other system, known as the *closed type*, the steam to be utilized is separated from the feed water by metallic walls, such as tubes of iron or brass after the fashion of a surface condenser. The water condensed in the operation of heating the feed is either drawn off or trapped. This type is used where the water is least contaminated, and is the one employed in U. S. naval vessels. Open heaters are used where the water is impure, especially where it contains lime or other scale-forming impurities, as they are provided with precipitating and filtering devices and deliver a comparatively pure boiler feed water.

The number of designs of heaters on the market is large. A description of several types is given in the text-book on "Boilers."

Feed Water Filters.—In order to remove oil or grease from the feed water before it enters the boilers and avoid its destructive effects, feed water filters are employed. The number of patented devices for filtration purposes in connection with feed water is exceedingly large and with varied claims for efficient qualities. In the U. S. Navy a filter tank is usually employed in connection with and as a part of the feed tank. The part devoted to the filter tank is usually divided by partitions into three chambers by means of perforated plates. In these chambers is placed the filtering material, consisting usually of some fibrous material, such as sponges, bagging, toweling, or a sea growth called "loofa." These materials collect the oil and impurities and before becoming clogged are removed, washed and cleaned or replaced by new. The discharge

from the air pumps lead into the filter tank through which the water flows to the feed tanks.

Grease Extractor.—There are many forms of apparatus for still further extracting grease from boiler feed water; the standard form used in the Navy is similar to a Macomb strainer, a basket or cartridge as it is called, being perforated so that the combined area of the perforations is about 30 times the sectional area of the feed line; the cartridge is covered with heavy linen toweling, is placed in the delivery feed line so water passes through the basket from the outside, the exit being from the open top of the basket, opposite to the direction of flow through a Macomb strainer. By-passes are fitted to permit continuous use of the line while the cartridge is removed for cleaning.

Auxiliary Condensers.—In order to provide for the exhaust from the numerous auxiliary engines on port service, and, when at sea, to guard against the unavoidable air leakages to the main condensers from so many distant auxiliaries, auxiliary condensers are employed. In the battleships and large cruisers, the number and size of the auxiliary engines render necessary the employment of two auxiliary condensers, one in each engine room.

The type universally used in the U. S. Navy is illustrated in Fig. 190 and is practically identical in construction with the main condensers described in Chap. IX. The exhaust steam enters at *A*, and is condensed by contact with tubes kept cold by water circulating through them, the condensed steam falls to the bottom and is drawn out at *B* by the air pump and sent to the feed tank. Circulating water is forced through *C* by the circulating pump into the lower half of the water chamber through the lower tubes and back through the upper half and through a pipe *D* to the overboard discharge valve.

The air and circulating pumps are of the piston pattern and combined, that is, a central steam cylinder is used, driving the air pump under one end of the condenser and the circulating pump at the other.

Evaporators.—While it is of the greatest importance that cylindrical boilers should be supplied with fresh water only, it is abso-

lutely necessary to the integrity of water-tube boilers that they should not use anything else. In order to supply the "make-up" fresh feed water rendered necessary by the leakage of steam and water through the joints, glands, drains, relief valves, whistles, etc., evaporators are installed in all vessels of the navy and mercantile marine.

The amount of water lost depends largely upon the design and workmanship of the various parts subjected to steam pressure, on

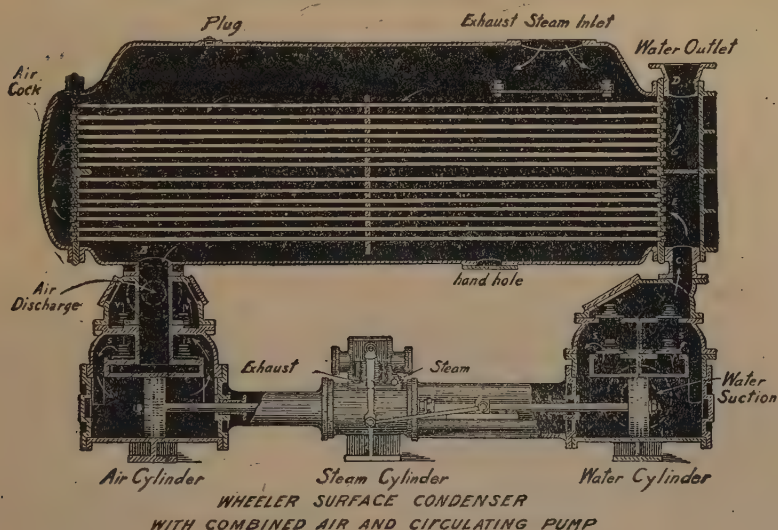


FIG. 190.

the appliances for minimizing waste and on the care of the officers in charge of the engineering department; in fact, the efficiency of that department can be largely determined by the amount of auxiliary feed required.

The amount of make-up feed required in vessels of war varies very considerably and is proportional to the length of steam and exhaust pipes, together with the number of engines connected with them. The British Committee on Naval Boilers recommended that not less than six tons per day be allowed per 1000 horse power, equivalent to about 3 per cent of the total feed, which is rather small for

the large number of auxiliaries employed, but probably based on maximum horse power, which is seldom used. Mr. Morrison estimates the loss of feed water by leakage at about eight tons per day for 1000 horse power for good passenger steamers, which is about 5 per cent of the total feed, assuming a total feed of 15 pounds per I. H. P. Also an estimate of ten tons per 1000 horse power for cargo boats, the assumption being that the leakage depends upon the care bestowed on the tightness of joints, glands, etc., and that less attention can be paid to them in the case of cargo boats. In addition to this for a large cruiser or battleship, with a complement of about 500 men, about 12 tons additional would be required per day. From the large quantities of fresh water required, the importance of keeping the evaporating and distilling plant at as high a degree of efficiency as possible can be readily seen.

The requisites for a good evaporator are that it must be efficient for the use it is intended and at the same time it must be simple in construction, requiring but little attention when at work, and capable of being cleaned quickly and with as little labor as possible.

There are many excellent evaporators in use for marine purposes of American and foreign design, but as they all act on the same principle, the only forms described will be those in general use in U. S. naval vessels; one designed by the Bureau of Steam Engineering, and the Reilly Multicoil Evaporator.

The Bureau type (Fig. 191) consists of a horizontal cylindrical or oval steel shell riveted with flat or rounded heads, braced to one another by through stay rods. The heating surface consists of a series of brass tubes forming a nest and expanded into one of the walls of the brass steam chambers *C* and *D* at each end, that at the front end being fitted with a diaphragm *E*. The opposite sides of these chambers are closed by bonnets *F* and *G*, for the removal and replacement of the heating tubes. The nest of tubes with the chambers is secured to the front head by means of a flange on the front chamber and stud bolts, and at the front and rear is supported by saddle plates, *H*. The whole nest with its chambers can be removed bodily, for cleaning and scaling whenever desired.

Operation.—Sea water is forced into the evaporator through the check valve at *I*, by means of the evaporator feed pump. The

primary steam is admitted into the upper part of the front chamber through pipe and valve *J*, passes through the upper tubes, and the water of condensation collects at the bottom and passes through the lower set of tubes beneath the diaphragm at front end, and is drained or trapped off to be sent to the feed tanks.

The steam generated from the heating of the sea water outside the tubes passes through the dry pipe *K* and out through the casting *M* with nozzles to the distiller, condenser, or second receiver, as desired, by appropriate pipes and valves, but it should not be led to the auxiliary exhaust pipe, the variation in pressure causing irregular evaporation with possible priming.

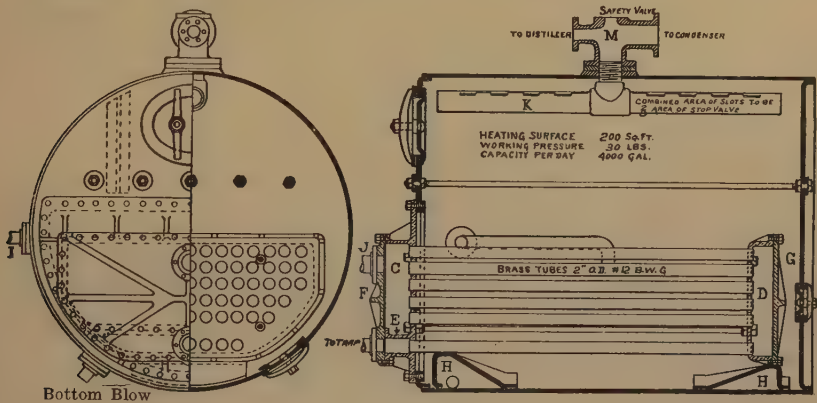


FIG. 191.

Steam is taken either directly from the boilers or from the receivers, and to maintain a regular and constant pressure a reducing valve should be fitted in addition to the stop valve on the evaporator coil steam pipe. As an evaporator is really a small boiler whose fuel is steam instead of coal, the mountings and attachments, such as steam gages, water gages, safety valve, bottom blow, etc., are fitted in the same manner as on the ordinary boiler.

Evaporators should be blown out at least four times a day; a convenient method for regularity is to blow down at the end of each watch. The density of water carried is from 3 to 4. For good working, the pressure from the secondary steam, or steam made from the salt water, is carried at from ten to fifteen pounds per gage.

When the evaporator is working on the main condenser, that is, when the secondary steam is being discharged into the condenser, the brine must be pumped out by means of the *brine pump*, and the feed will flow in. In this case the vapor is formed at a much lower temperature than when there is a pressure greater than that of the atmosphere in the evaporator.

The following precautions should be taken: The prevention of priming by the proper determination and maintenance of water level, the amount of steam supplied to the coil or tubes should be regulated to the amount of water evaporated, so as to keep the pressure of the secondary steam constant, and the steam coil drains should be opened sufficiently to drain hot water only. Owing to its high evaporating power, an evaporator will prime unless carefully handled. The water level must not be carried too high, the best results being obtained when it is just below the upper surface of the tubes, as in this case the bubbles of water carried above the surface with the steam are evaporated by the exposed heating surface.

The nest of tubes should be hauled out at periodic intervals (by breaking the joint on the evaporator head) and thoroughly scaled, cleaned and replaced, the operation taking from six to eight hours. Usually a spare nest of tubes is carried, so that one may be cleaned at leisure, while the spare is put in use. Zinc protective slabs are fitted in an evaporator similar to those fitted in the boiler.

Reilly Multicoil Navy Type Evaporators are constructed with vertical or horizontal shells, depending upon the capacity. In both types the arrangement of coils and internal parts is identical.

Fig. 192 shows a horizontal shell evaporator of the single coil type; larger evaporators of this make, but with two coils side by side in one shell, are installed on some of our recent battleships.

Shells are of open hearth boiler and flange steel; doors and frames of flange steel.

Stiffeners on heads of horizontal shells are specially designed to eliminate stay rods.

The manifolds to which coils are attached are of bronze; coils are of double refined and annealed copper tubing rolled to shape without filling, in order to secure a uniform thickness of metal throughout.



FIG. 192.

Coil ends are equipped with patented quick detachable flanged joints, copper faced and free from brazing, making a metal-to-metal bolted joint.

Baffles of a patented design are arranged so as to increase circulation and reduce priming.

The handling gear for removing coils and manifolds consists of composition rollers on the inside of the evaporator, and a jack roller attached to the door.

The manifolds are securely bolted to the door, the coils being bolted to the steam and drain manifolds, thus permitting the entire nest of coils to be removed upon releasing the door. The weight of the door is taken off the studs by means of the jack roller. A sight hole is provided in the center of the door to permit of easy inspection. Moreover, through this opening a stream of cold water may be played on the coils while they are under pressure, cracking off a large percentage of scale by sudden expansion and contraction, for which this type of coil is particularly adapted.

Zinc plates are placed inside to reduce galvanic action.

Multiple-Effect Evaporating Plants.—Large evaporating plants are fitted with only the first evaporator, in a set of three, using steam from the boiler, the steam produced passing to the tubes of a second evaporator and the steam produced in the second to the tubes of a third, the third evaporator discharging steam to the condenser or distiller as before. The object of this arrangement is to obtain the maximum amount of water per pound of coal used.

From a series of experiments on evaporation conducted by Mr. D. B. Morrison, the investigations show that water may be obtained from an evaporator for a small expenditure of coal, or by a very wasteful consumption, depending on the arrangements adopted. He found that taking the primary steam from the second receiver is more economical than a direct connection to the boiler, but the practical considerations are in favor of the latter. His results also showed that sending the secondary steam to the condenser, though a cheap method as regards the cost of fitting up the apparatus, was distinctly the most uneconomical that could be adopted. It may be necessary in port, but at sea the feed water in the hot well will readily absorb all the heat in the secondary steam, thus returning the heat to the boiler instead of sending it overboard.

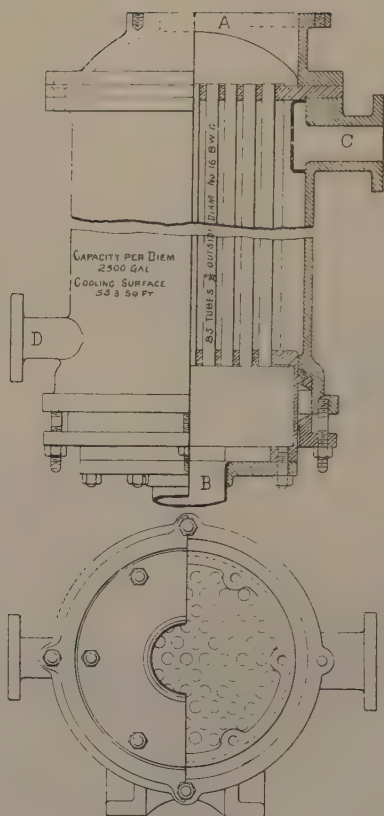


FIG. 193.

The results obtained by Mr. Morrison in comparing the relative economy of taking the primary steam from the boiler and from the receiver, and of sending the secondary steam to condenser and to hot well show great possible economy. The cost of obtaining fresh water by means of an evaporator under different conditions is an important consideration when viewed from a commercial side, but the most economical methods are not practicable in cruising ships for making drinking water.

Distillers.—If the fresh water from the evaporators is required for drinking purposes, the secondary steam is passed through a distiller, condensed and filtered before it is pumped to the ship's tanks. The distiller used in the U. S. Navy is of Bureau design, made of various sizes to suit different vessels, and is shown in Fig. 193. The shell is of cast iron and tubes and tube sheet of composition, the parts in contact with the fresh water being tinned. The shell is placed upright when possible, the end *A* at the top. The tubes are expanded into the tube sheets, the lower tube sheet being arranged to slide in a stuffing box to allow for the expansion of the tubes. The condensing water enters at *B* and flows out at *A*. Steam from the evaporator enters at *C* and the water of condensation leaves at *D*. The perforated diaphragm under *C* distributes the entering steam over a wider area and prevents injury to the tubes at entrance. The circulating water through the distiller is taken from the sea and forced through the pump by the *distiller circulating pump* overboard or through the flushing pipe at will. The condensed steam or fresh water falls into a small reservoir, from which it is taken by the *distiller fresh water pump* and sent forward to the ship's tanks. The distiller circulating water can also be taken from the fire main by suitable connections in case of necessary overhauling or repairs to the distiller circulating pump.

Turning Engine and Gear.—In order to move the main engines when not in use and to facilitate adjustments and repairs, small double engines are fitted in each engine room. This engine is made to drive, by worm gearing, a worm wheel on the crank shaft, and is arranged so as to be readily thrown in and out of connection with the crank shaft. A differential valve gear is fitted to the engine so that the main engine may be moved in either direction. The turning engine shaft is also fitted with a ratchet and wrench, or any

approved hand turning gear, in case the turning engine is disabled. Before admitting steam to the main engines care must be taken to see that the turning engine is disconnected.

Steam Steering Gear.—For large vessels the employment of some

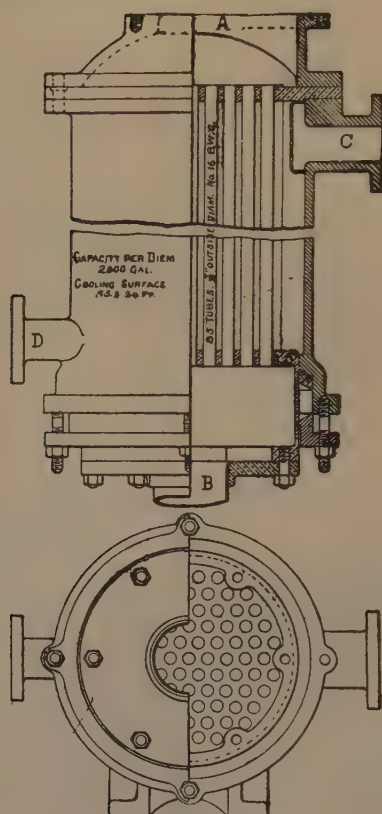


FIG. 193.

power other than hand is needed. With steam one man is enabled, while the ship is running at full speed, to throw the rudder over from one extreme position to the other with but little exertion and complete safety. In the design of such a gear the requisites are safety and reliability in operation and simplicity in construction and arrangement.

The Williamson steering engine, Plates 31 and 32, generally used in cruisers and battleships in our navy, is fitted with a differential reversing valve, described in Chap. V. Plates 31 and 32 show a general plan and elevation of the complete gear which consists of a two-cylinder engine *B-B* driving a crank shaft *H*, which communicates with the tiller shaft by suitable gearing explained below, and which reduces the speed of this shaft and increases the turning moment, so as to overcome the resistance of the rudder at high speeds. The principles underlying the operation of this steering engine are the same as those used in later engines, and the detailed description will be confined to this type, which is still in use on many ships. Plate 35 shows the general arrangement of one of the latest installations; the principal improvement is the change in location of the steering engine, which is now placed adjacent to the main engine room instead of in the tiller room. Shafting transmits the power to the right and left hand threaded screw. The mechanical details of the follow-up gear, and of the transmission of the power of the after telemotor to the automatic shaft *E* have been modified in recent installations, the principles remaining unchanged.

By means of the differential reversing valve (*A* in Plate 33) the engine moves when the small steering wheel is moved and stops when the movement of this wheel is discontinued by the operator.

The gear is arranged as follows (See Plates 31, 32 and 33) :

E is the controlling or automatic shaft in communication with the steering wheels at different parts of the ship.

H, steering engine crank shaft carrying a small spur wheel *T*, which gears into a large spur wheel *S*, fitted loosely on tiller shaft *C*, but to which it can be secured by the clutch *M*. Aft of the above large spur wheel (to the *left* in Plate 33) is a small pinion gearing into a pinion *Q*, on the controlling shaft *E*, through the idler *R*.

The tiller shaft extends from the steering wheel *P* aft to the bearing just forward of the rudder head. The after part is threaded with a right and left-hand screw thread, on which work the driving nuts *G*, kept in line by side rod guides. The driving nuts move in opposite directions and are each connected to an end of a yoke or crosshead on the rudder head, by means of connecting rods.

The arrangement by which the engine moves in the direction turned by the steering wheels, in different parts of the ship, and stops when the steering wheel stops, is accomplished as follows:

Leaving aside for the present the gearing and connections to the operating wheels outside the tiller room, and referring to Plate 33, we will consider the action of the reversing valve as operated by the emergency steering wheel in the tiller room. On the controller or automatic shaft *E* there is a sleeve *L*, with internal threads fitting into corresponding threads on the shaft *E*. At the end of this automatic shaft there is a steering wheel *I*. A groove is cut in the sleeve *L*, in which is fitted a loose collar or strap which in connection with a series of bell cranks *F* moves the reversing valve in one direction or the other according to the way the wheel *I* is turned, and the engine moves accordingly. The engine will stop when movement of the steering wheel ceases, for the pinion *Q* secured to the sleeve gears into the spur wheel on the tiller shaft through the idler *R*, which shaft is in connection with the crank shaft of the engine by another set of gear wheels, *S* and *T*. On turning the steering wheel *I* in one direction, that shown by the arrow on *E*, for instance, the nut *L* (being prevented from turning by *R*) is drawn to the right, the valve *A* is opened and the engine starts; upon starting, the shaft *C* turns and in so doing not only moves the rudder, but, through *R*, turns *L* in the same direction *E* was turned, thus following up the latter, so that, when the operator ceases to turn *I*, the small movement of the engine turns *L* back, closing the reversing valve, and the supply of steam is shut off from the cylinder.

Consequently the steering engine works only while the steering wheel is being moved, and when this wheel is not moved further, the engine, and with it the rudder, remains fixed in whatever position it may be placed, and remains there until the operator again moves the steering wheel.

To prevent the steering wheel from being turned too far, stops are arranged by means of a traveling nut *U*, with a projecting lug engaging in the slot in the bearing *V*. A connection is made through this lug to a pointer on a graduated rudder telltale, forward of the steering wheel.

The above arrangement illustrates the method of working the rudder from the tiller room. It can also be worked under steam by steering wheels at the following places, viz.: The conning tower, the central station, and an easily dismounted portable station for use in piloting on top of the conning tower, or other temporary bridge station.

The transmission of the motion of the steering wheel at these points to the automatic shaft *E* on engine is made either by shafting and bevel wheels or wire rope wound on barrels as in the ordinary hand steering gear. In recent ships a telemotor for all the forward stations is installed to work the controlling shaft in the tiller room, while the wire rope transmission is retained as an emergency gear in case of break-down.

Motion is transmitted with the telemotor through water pressure in small pipes from a transmitting cylinder placed below the protective deck (Plate 34). The steering wheel is connected by a vertical shaft and gearing to the piston rod of the transmitting water cylinder, and by means of the piping from this cylinder any motion of the transmitting piston produces similar motion to the piston of another receiving water cylinder in the tiller room, which is in connection by a combination of simple levers with the controlling or automatic shaft *E*, by which the reversing shaft gives the desired motion to the engine as previously explained.

In addition to the connections for steam steering, provision can be made for steering by hand in case of disablement of the steam steering gear, by means of the hand steering wheel *P* in the tiller room.

In several ships of the navy hydraulic power has been employed instead of steam power and with fairly good results. Electric power has been satisfactorily used, and now seems proving itself equal in all respects to steam.

The half-tone, Fig. 194, was taken from the working model of the steam steering gear in the Model Room of the Engineering Department. It is a representation of the earlier type of this gear in which motion to the controlling shaft in the tiller room is transmitted by shafts and gearing, but is capable of nearly all the changes in gearing given by the improved type.



FIG. 194.

The following table shows the clutches which must be connected or disconnected in shifting from steam to hand, or shifting from one steering station to another in the use of this model:

TABLE VIII.

CONNECTIONS FOR WILLIAMSON'S STEERING GEAR (IN MODEL ROOM).

To steer from.	Method of steering.	Connections in.	Connections out.
Bridge.	Forward station.		2, 3, 6, 7, 8 & 11.
	Steam.	1, 4, 5, 9 & 10.	
Upper deck; Upper deck.	After station.		1, 3, 6, 7, 8 & 11.
	Steam.	2, 4, 5, 9 & 10.	
	Hand power.	2, 3, 5, 9 & 10.	1, 4, 6, 7, 8 & 11.
On engine. At engine.	Station in steering engine room.		2, 3, 6, 7, 8 & 11.
	Steam.	4, 5, 9 & 10.	
	Hand power.	6, 7, 8, 9 & 10.	5 & 11.
Tiller room.	Station in tiller room.		9 & 10.
	Block and tackle.	11.	

Electric Steering Gear.—Recent successes with electric steering indicate an early adoption of that type in place of steam. Plate 35 shows the general arrangement of steering gear now prevalent in the navy, and an electric rudder-motor, as well as the usual steam engine, is there shown. Either steam or motor may be used—a clutch being provided in the transmission shaft to cut out the steam engine. The rudder-motor is geared to the transmission shaft by spur reduction gearing.

The details of the motor and controller belong to electrical engineering. There are two principal systems of controlling the motor, however; one is the *follow-up* system, in which the steering wheel is put over and the motor, upon starting and moving the rudder to the desired angle, stops itself just as the steam engine does. In this system the operation of steering appears to the helmsman quite like steering with steam. The controller is located in or near the steering gear room and is connected to the usual form of steering stand through shafting, or tiller ropes, etc., as with steam gear. The *non-follow-up* system requires a special type of steering stand, and the helmsman finds the operation of steering quite different from the follow-up gear. This system requires no mechanical connections with the bridge whatever—electrical con-

nection only is needed. A reversing switch in the steering stand, operated by a single lever, starts the motor slowly by a small movement; greater movement of the lever accelerates the motor; and throwing the lever in reverse direction reverses the motor. As long as the lever is held in position for moving the rudder-motor the rudder continues to move—there is no “follow-up”—until the limit switch is reached at “hard over,” where the motor is automatically stopped. Bringing the lever to the central position stops the motor; a powerful electric disc brake on the armature shaft assists in bringing the motor promptly to rest when the steering lever is placed in the central position.

The steering lever is so arranged that upon being released it instantly returns to the central position, thus stopping the motor.

In the non-follow-up system a rudder indicator is depended upon solely to show the rudder angle.

The latter system is the simpler and better one of the two; the follow-up type of electric gear may be considered as abandoned.

Ash Hoist Engine.—The ash-hoisting engine found in American naval vessels is illustrated in Fig. 195, and consists of two cylinders with overhung cranks driving the hoisting drum; each engine is fitted with a single eccentric to work its slide valve.

The ash bucket is raised or lowered by working the engine, by the hand wheel on the left, which is kept moving in the direction it is desired that the drum should revolve, thus keeping open the differential reversing valve (similar to the one shown on page 110) which controls the admission and direction of the steam for the engine. When the movement of the hand wheel ceases, the reversing valve closes and stops the engine.

The actuating gear is almost identical with the automatic controlling gear just described for the steering engine, and differs only in details.

In Fig. 195, showing the complete ash hoist engine, *A-A* are the single eccentrics actuating the main valve on each engine, *B* a pinion fixed to the drum and gearing into a spur wheel *C* cast on a sleeve *L* with an internal thread to correspond to the external thread on the shaft *E*. The arrangement and manner of action is exactly similar to that of the sleeve *L* and the shaft *E* on the steam steering

engine shown in Plate 33. A groove is cut in the sleeve fitted with a loose collar *G*, which through a bell crank *F* moves the reversing valve by means of the valve stem *H* when the spur wheel *C* moves to the right or left. Turning the starting or hand wheel *D* clockwise the spur wheel *C* moves to the left, the differential valve moves downward and the engine turns, moving the drum shaft anti-clockwise, the spur wheel *C* will then have a motion clockwise, pushing the sleeve to the right, closing the differential valve, with the result that if the hand wheel is stopped, the engine stops, or if the hand wheel is turned in the opposite direction, a reverse movement to that given above takes place.

The screw thread shown in Fig. 195, on the automatic shaft, carries a nut *J* which is prevented from turning, but is allowed sufficient lateral motion between stops to give the proper amount of hoist. On bringing up on the stops, it prevents the bucket being raised or lowered too far, and thus avoids danger of breaking the rope.

See's Ash Ejector.—This apparatus (Fig. 196) is fitted in many passenger steamers to obviate the disagreeable features in hoisting ashes on deck. *A* is a hopper or trough in the fire room, *D* a connecting pipe or chute leading up through the side of the ship, slightly above the water line. A discharge pipe from a pump is led to the lower part of the hopper; the pipe having a contracted nozzle, the water issues with a high velocity. Starting with the cover closed on the hopper, the water issues from the upper outboard end of the pipe with great velocity. If now the cover is removed and the ashes thrown in, the stream scours the ashes through the pipe and ejects them free of the ship's side.

The above is a simple ejector. Another type, called the compound ejector, has an air chamber which maintains a steadier, uninterrupted flow of water, with less liability to choking.

Ash Expeller.—Some of our recent ships are fitted with *ash expellers*. The expeller is a vertical tube of about 7 inches diameter, connecting a hopper located above the fire room floor with the sea beneath the outer skin of the ship. The lower part of the tube is surrounded by an annular chamber contracted at the lower end to form an ejector nozzle of the same diameter as the upper part of the

tube; a pump connection to the annular chamber is provided above the inner bottom. There are two large gate valves in the tube, the upper one at the bottom of the hopper, the second one just above the pump delivery to the annular chamber. Both these gates cannot be opened at the same time, thus obviating danger of flooding the ship. A third gate near the outer skin of the ship is provided, which can

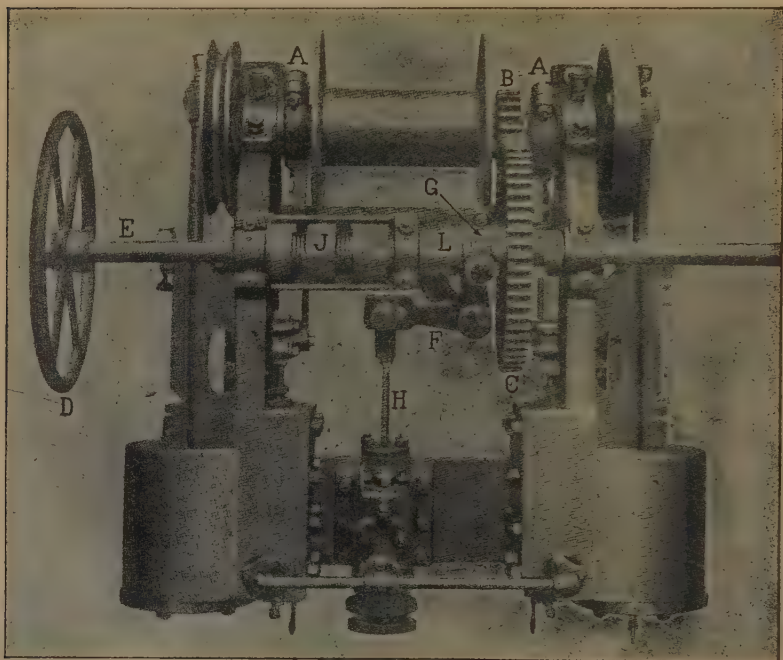


FIG. 195.

be closed to permit overhauling the apparatus without docking the ship, but does not concern the operation of the expeller.

The hopper has an ordinary cover secured by dogs with which to keep it closed when not in use. Starting with the upper gate open and the second closed: ashes are shoveled into the hopper, the pump is started on the annular chamber, supplying sufficient water pressure so that the kinetic energy of the jet issuing downward is con-

siderably in excess of the static pressure at the mouth of the tube due to the water-line head, preventing any water entering, and even affording a slight vacuum in the upper part of the tube to accelerate the descent of ashes. When the hopper is filled and the water pressure is on, by simply opening the second gate and closing the upper one the ashes in the tube descend by gravity assisted by the vacuum mentioned above and are forced out by the water jet.

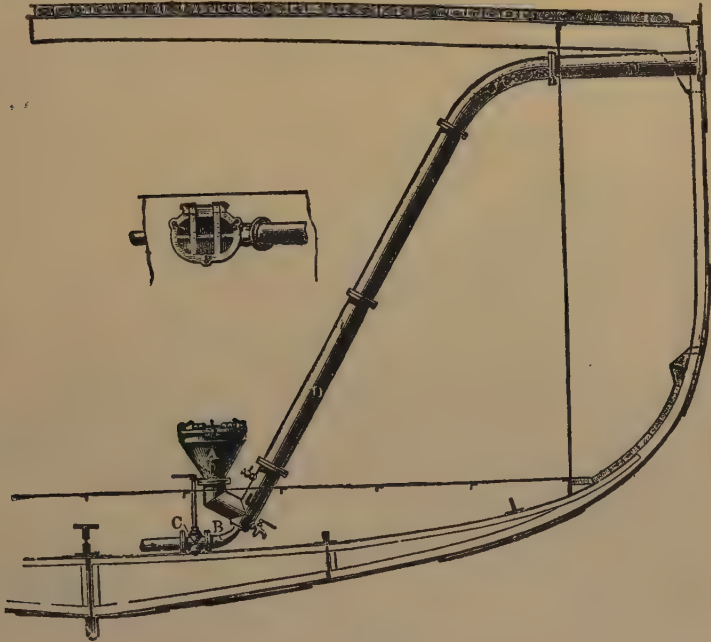


FIG. 196.

Steam Capstan Windlass.—The steam capstan windlass shown in Fig. 197 is of the horizontal type, and drives the capstan on the deck above through bevel gears, thus permitting the windlass to be operated by capstan bars if necessary. The windlass and capstan are driven by a double vertical engine whose cranks are attached to a shaft having an endless screw gearing with a worm-wheel on the windlass shaft. The engines have one eccentric each and are fitted with a reversing valve (Fig. 51) by which they may be reversed at will.

The windlass shaft *A* is arranged to carry four wildcats or chain grabs *B*. These wildcats are loosely mounted on the windlass shaft and are controlled by friction band brakes and are adapted to be locked to rigid driving heads keyed on the shaft.

The projecting rim of the wildcat, for receiving the band of the friction brake, has ratchet-shaped thrust blocks cast on its inner

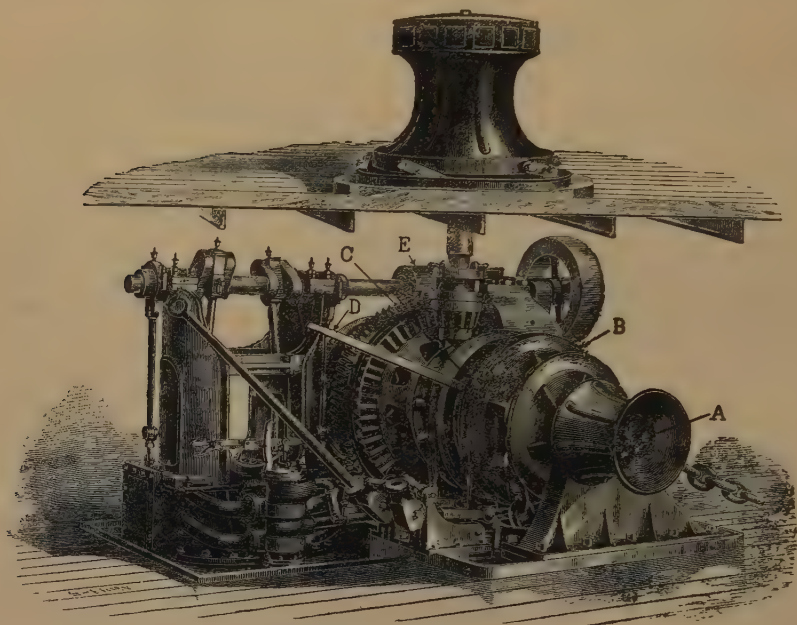


FIG. 197.

side, against which locking pawls are brought in contact or drawn away from same, as it is desired to lock or unlock the wildcat.

The locking and unlocking of the wildcat is accomplished by the application of a ring *C* loosely mounted upon the driving head and having cam-shaped grooves or slots into which a pin, secured to the locking pawl, enters, so that the turning of the ring on the driving head causes the locking pawl to be either thrown out or drawn in, thus locking or unlocking the wildcat. The windlass is provided with locking levers *D* for turning the locking ring.

The worm gear *E* of the windlass is loose on a driving head of the windlass shaft, but can be connected with the same by inserting a key block.

The bevel gear on the windlass shaft is provided with a ratchet wheel into which properly arranged pawls from the pawl bitt enter,

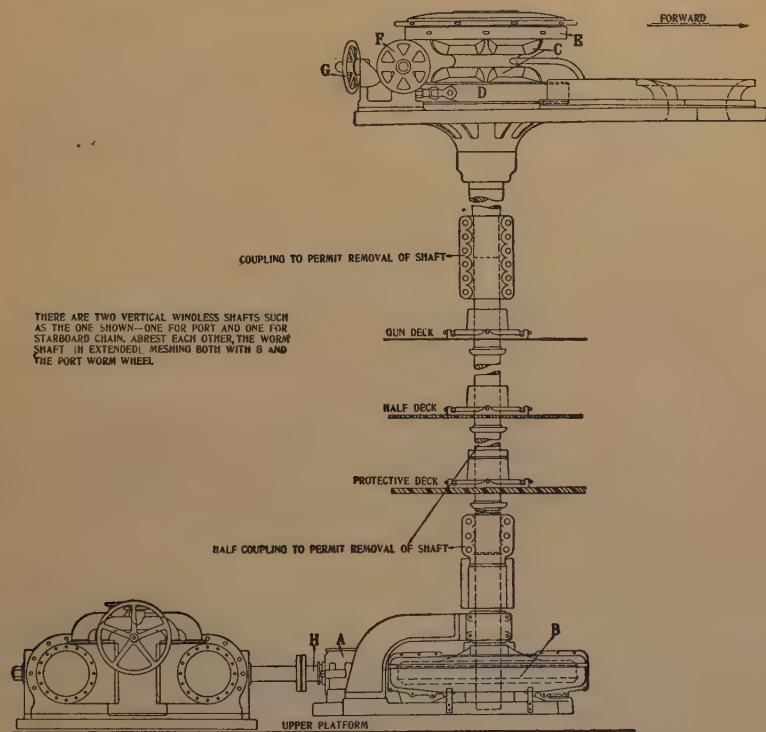


FIG. 198.

so that the shaft and driving heads, secured to the same, as also the wildcats when secured to the heads, will be held firmly in any position desired, and so hold the chains and prevent paying out of same.

The thrust of the worm is taken by hardened steel and bronze thrust washers, and the wear of the thrust bearing and washers is provided for by the adjustment of steel screw bolts.

The *Vertical Steam Windlass* is displacing the capstan just described. In this type the engine, which may be either horizontal or vertical, is located on one of the lower decks. The crank shaft extended has a thrust bearing, and beyond that ends in a worm driving a pair of horizontal worm wheels on two vertical windlass

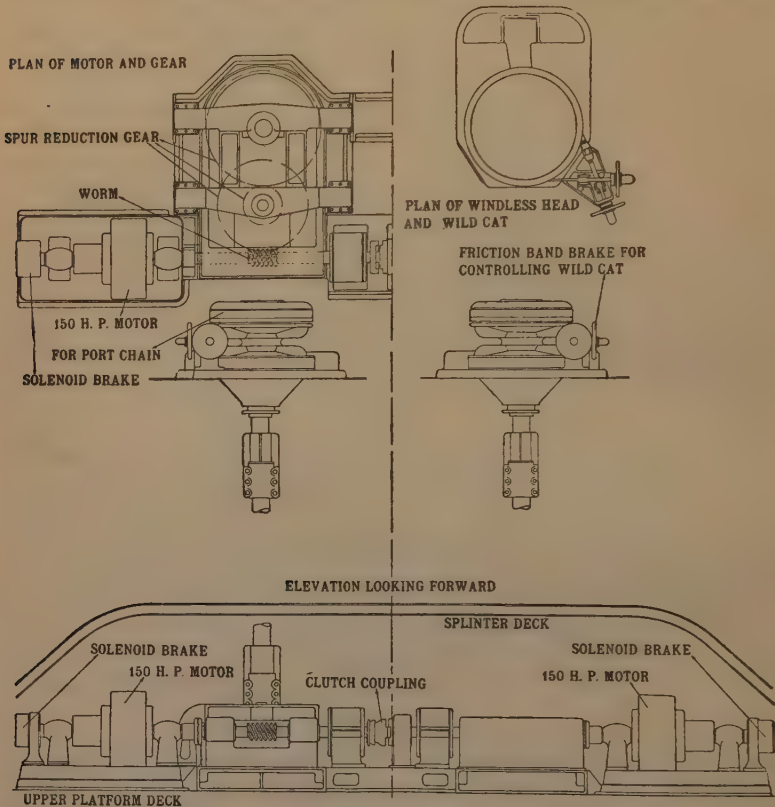


FIG. 199.

shafts; the latter, passing through the various decks with suitable stuffing boxes, drive two horizontal wildcats on the forecastle deck, which are controlled similarly as described for the capstan type. The engine is double and usually is fitted with double eccentrics and link reversing gear instead of the differential reversing valve. Figure 198 shows one view of a recent vertical design. *A* is the thrust

bearing, *B* one of the worm wheels into which the shaft *H* (ending in a worm shaft) gears, *C* is the wildcat which can be held firmly by the friction hand brake *D*, or can be locked to the vertical shaft by the locking ring *E* at will, as previously described for the horizontal type. *F* is the hand wheel for operating the friction brake. *G* is a similar brake wheel for the port windlass.

The general arrangement of another recent vertical design, operated by electricity, is shown in Fig. 199. The sketch explains itself.

Air Compressing Machinery.—Air compressors are fitted to ships for charging and launching torpedoes, also in the Engineering Department a compressor is installed in some of the later designed armored cruisers and battleships for use in running pneumatic tools and removing soot from the boiler tubes.

There are various arrangements in use in which the compression is done in several stages by the use of cylinders with successively decreasing diameters. In the U. S. Navy the types in general use are the Westinghouse steam driven, Ingersoll-Rand electric driven, and Platt Iron Works, either steam or electric driven. The first mentioned is generally used in the Engineer Department on our ships and will be described in detail.

Westinghouse Steam-Driven Compressor.—The compressor consists primarily of an air cylinder, in which the air drawn from the atmosphere is compressed, a steam cylinder, located above the air cylinder, the two being connected by a suitable center piece, steam and air pistons mounted on a common piston rod, and a valve motion controlling steam admission and exhaust. The compressor is double acting, steam being admitted alternately on either side of the steam piston which, being directly connected with the air piston, causes both to move up and down.

On the upward stroke of the air piston the air above it is compressed and discharged into the main reservoir, while the space below is filled with air drawn from the atmosphere. On the downward stroke this operation is reversed.

A *compressor governor* is placed in the steam supply pipe between the lubricator and the compressor, its function being to start and stop the compressor automatically within predetermined pressure limits.

Views of the compressor with steam and air cylinders and valve mechanism in section are shown in Figs. 200 and 201. Figs. 202

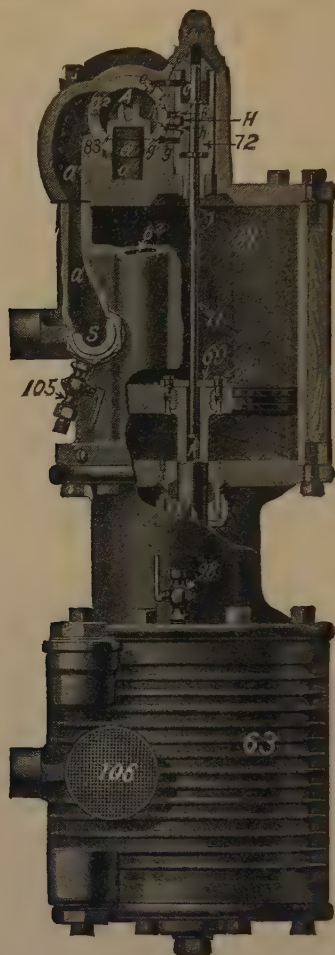


FIG. 200.

SECTION OF COMPRESSOR THROUGH REVERSING VALVE

and 203 are distorted or “diagrammatic” illustrations designed to show as clearly as possible the connections of the various ports and passages, but not the actual construction of the parts.

Steam End.—Considering first the steam end of the compressor, and referring to Figs. 200, 201, 202 and 203, steam from the supply

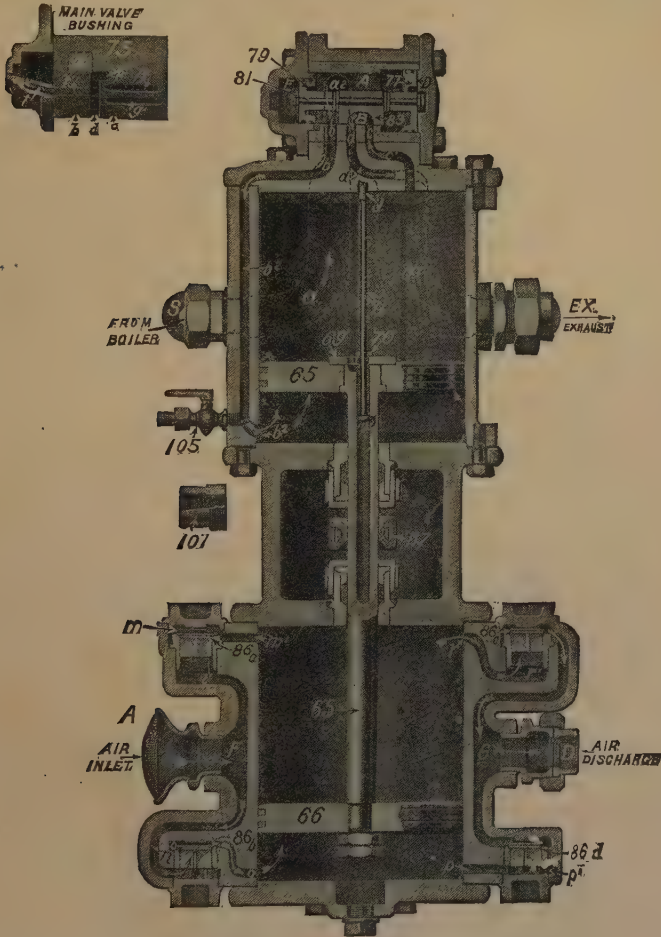


FIG. 201.

SECTION OF COMPRESSOR THROUGH MAIN VALVE

enters at the connection marked "From Boiler," Fig. 201, or "Steam Inlet," Figs. 202 and 203, and flows through the passageways a , a^2 to the chamber A , above the main valve 83 and between the pistons

77 and 79, and through passage *e* to chamber *C*, in which is reversing valve 72. The supply and exhaust of steam to and from the steam cylinder is controlled by the main valve 83, which is a "D" type of slide valve. It is operated by the two pistons 77 and 79, of unequal diameters and connected by the stem 81. The movement of these two pistons and the main valve is controlled by the reversing valve 72, which is in turn operated by the main steam piston 65, by means of the reversing rod 71, and the reversing plate 69. As will be seen from the following description, the duty of the reversing valve 72 is to alternately admit steam to or discharge it from chamber *D*, at the right of piston 77, thus alternately balancing or unbalancing this piston. The reversing valve is operated by the reversing rod 71. This rod is alternately moved up and down by reversing plate 69, which engages reversing shoulder *j*, Fig. 200, on the upward stroke of the steam piston and button *k*, at the end of the rod, on the downward stroke.

Chambers *A* and *C* are always in free communication with each other and with the steam inlet through port *e*, *e'* as shown in the cuts. Live steam is therefore always present in these chambers, *A* and *C*. Chamber *E*, at the left of small piston 79, is always open to the exhaust passage *d*, through the ports *t* and *t'*, shown in the main valve bushing, Fig. 200, and diagrammatically in Figs. 201 and 202. Exhaust steam, practically at atmospheric pressure, is therefore always present in chamber *E*.

A *balancing port*, *s*, Figs. 202 and 203 (shown also in Fig. 200), runs diagonally to the right in the reversing valve cap nut, and meets a groove down the outside of the reversing valve bush, where it enters the upper end of the cylinder through a small port in the head. The object of this is to assure the same pressure above as below the reversing rod, whether there is live or exhaust steam in the upper end of the cylinder, thus balancing it so far as steam pressure is concerned.

When reversing slide valve 72 is in its lower position, as shown in Figs. 200 and 202, chamber *D* is connected (through ports *h*, *h'*, reversing valve exhaust cavity *H*, Fig. 200, and ports *f* and *f'*) with main exhaust passage *d*, *d'*, *d*,² and there is, therefore, only atmospheric pressure at the right of piston 77.

Therefore, as chamber *E*, at the left of piston 79, and chamber *D*, at the right of piston 77, are then both connected to the exhaust, as already explained, the pressure of the steam in chamber *A* has driven the larger piston 77 to the right, and it has pulled the smaller piston, 79, and the main valve, 83, with it to the position shown in Figs. 201 and 202. The main valve, 83, is then admitting steam below piston 65, through port *b*, *b'* *b*². Piston 65 is thereby forced upward, and the steam above piston 65 passes through port *c'*, *c*, exhaust cavity *B* of main valve 83, port *d*, and passage *d'*, *d*² to connection *Ex*, at which point it leaves the compressor and discharges through the exhaust pipe into the atmosphere.

When piston 65 reaches the upper end of its stroke, reversing plate 69 strikes shoulder *j* on rod 71, forcing it and reversing slide valve 72 upward sufficiently to open port *g* (Figs. 200 and 203). Steam from chamber *C* then enters chamber *D* through port *g* and port *g'* of the bushing. The pressures upon the two sides of piston 77 are thus equalized or balanced. Considering piston 79, the conditions are different. Chamber *E*, as already stated, is always open to the exhaust. As piston 77 is now balanced, the steam pressure in chamber *A* forces piston 79 to the left, drawing with it piston 77 and main valve 83, to position shown in Fig. 203.

With main valve 83 in this position, steam is admitted from chamber *A*, through port *c*, *c'* above piston 65, forcing it down; at the same time the steam below this piston is exhausted to the atmosphere through port *b*², *b'*, *b*, exhaust cavity *B* in the main valve, port *d*², *d'*, *d*, and the exhaust pipe connected at *Ex*.

When piston 65 reaches the *lower* end of its stroke, reversing plate 69 engages reversing button *k*, and draws rod 71 and reversing valve 72 down to the positions shown in Figs. 201 and 202, and one complete cycle (two single strokes) of the steam end of the compressor has been described.

Air End.—The movement of steam piston 65 is imparted to air piston 66 by means of the piston rod 65. As the air piston 66 is raised, the air above it is compressed, and air from the atmosphere is drawn in beneath it; the reverse is true in the downward stroke.

On the upward stroke of piston 66, the air being compressed above it is prevented from discharging back into the atmosphere by upper

inlet valve 86a. As soon as the pressure in ports r, r' below upper discharge valve 86c becomes greater than the main reservoir pressure above it, the discharge valve 86c is lifted from its seat. The

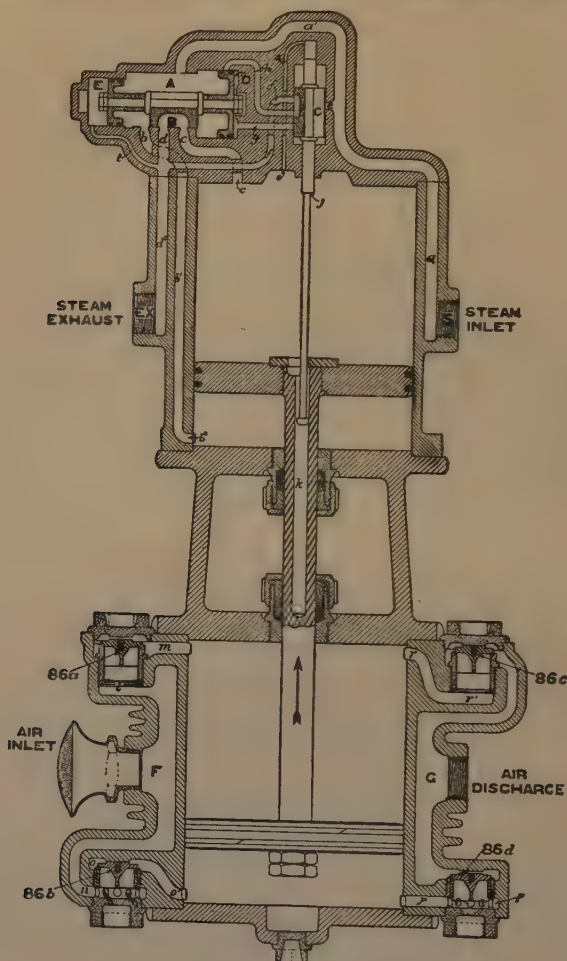


FIG. 202.

DIAGRAMMATIC VIEW, UP STROKE

air then flows past this valve down through chamber G , out at the "Air Discharge" and through the discharge pipe into the main reservoir.

The upward movement of the air piston produces a partial suction or vacuum in the portion of the cylinder below it. The air

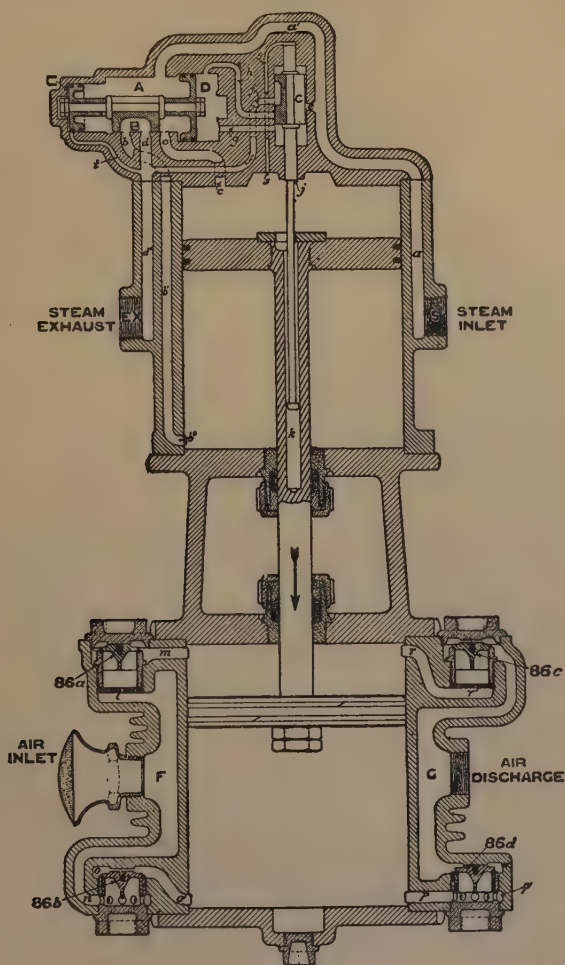


FIG. 203.

DIAGRAMMATIC VIEW, DOWN STROKE

pressure below piston 66 and on top of the lower left-hand inlet valve 86b becomes, therefore, less than that of the atmosphere in port *n* underneath this valve. Atmospheric pressure, therefore, raises valve

86b from its seat, and atmospheric air is drawn through strainer 106, at the "Air Inlet," into chamber *F*, and port *n*, below the inlet valve 86b, thence past that valve and through ports *o* and *o'* into the lower end of the air cylinder, filling same. Air cannot enter this part of the cylinder by flowing back from the reservoir through passages *D* and *G* and lower discharge valve 86d, since this valve is held to its seat by the main reservoir pressure above it. The lower inlet valve 86b seats by its own weight as soon as the up stroke of the air piston 66 is completed.

On the downward stroke of the compressor, the effect just described is reversed, the air below piston 66 being compressed and forced out through ports *p* and *p'*, past lower discharge valve 86d and through chamber *G* and the air discharge pipe into the main reservoir. At the same time air is being drawn in from the atmosphere through the "Air Inlet" through chamber *F* and port *l'*, upper inlet valve 86a and ports *m* and *m'* into the upper end of the air cylinder above the air piston 66.

The inlet and discharge valves of the 11-inch, $9\frac{1}{2}$ -inch and new pattern 8-inch compressors should each have a lift of $\frac{3}{32}$ of an inch.

The governor should be located in the steam supply pipe *between the lubricator connection and the compressor* in order to insure its receiving the necessary lubrication. The lubricator connection consists of a tee, the side outlet of which connects to the lubricator. In two-compressor installation the governor should be located in the main steam supply pipe between the lubricator fitting and the steam branch pipe leading to each compressor.

Starting and Running.—The drain cocks (105, Figs. 200 and 201) are placed at the lowest points of the steam passages *a* and *b*,² as shown, for the purpose of draining water from steam condensed after the compressor stops. They should always be left open when the compressor stands idle for any length of time. These drain cocks are provided with suitable union fittings so that drain pipes may be connected to them if desired.

In starting a compressor, always run it slowly until it becomes warm, permitting the condensed steam to escape through the drain cocks and the exhaust, until there is sufficient pressure in the main

reservoir (25 to 30 lbs.) to provide an air cushion. Then close drain cocks and open the steam (throttle) valve sufficiently to run the compressor at the normal speed. Never run the compressor faster than is necessary to do the work required. Racing or running at excessive speeds should not be allowed. The compressor governor automatically controls the starting and stopping of the compressor.

Lubrication.—Steam Cylinders.—The *steam cylinder lubricator* should not be started until all condensation has escaped from the compressor and the drain cocks closed. After closing the drain cocks start the lubricator to feed in ten or fifteen drops of oil as rapidly as possible, then regulate the feed to about one or two drops per minute *for one compressor*. No definite amount can be specified, as the amount of lubrication required depends on the work the compressor has to do, the quality of the steam, condition of compressor and so on. Keep the lubricator feeding while the compressor is running.

Lubrication.—Air Cylinder.—To oil the *air cylinder*, open its oil cock (98, Fig. 200), and blow out all dirt, close and fill it with *valve oil*, and on the *down stroke* of the piston open the cup to allow the oil to be drawn into the cylinder, closing the cup before the beginning of the up stroke. This is most easily done when the speed is moderate and the air pressure low. Valve oil only should be used in the air cylinder. A lighter oil will not last and is dangerous. A heavier oil very soon clogs and restricts the air passages, causing the compressor to heat unduly and compress air slowly; valve oil gives the best performance. Judgment should determine the amount for both the air and steam cylinders, it being remembered that the lack of a little oil when needed may result in many dollars' worth of damage to the compressor.

A *swab*, well oiled, is essential on the piston rod.

The sight feed lubricator gives complete and convenient control of air cylinder lubrication, so that the minimum amount and proper quality of oil required may be supplied at regular intervals. In order to prevent compressed air from entering the oil delivery pipe between the sight feed fitting and the air cylinder, a ball check valve connection is screwed into the air cylinder. No trap should exist in the oil delivery pipe between the sight feed fitting and the air cylinder.

To Stop the Compressor.—(1) Close the feed and steam valves on the sight feed lubricator; (2) then close the steam (throttle) valve; (3) and open all drain cocks on the compressor. Keep the steam valve closed and the drain cocks open when the compressor is not working. The main reservoir drain cocks should also be left open when the compressor is stopped for any length of time.

DISORDERS.

CAUSES AND REMEDIES.

Compressor Refuses to Start.—Cause: insufficient oil, from scant or no feed or working water; worn main-piston rings; or rust having accumulated during time compressor has lain idle. Remedy: shut off steam, take off cap nut, put in about a tablespoonful of valve oil (not too much), let the oil soak down for one or two minutes, and then turn on steam quickly. In many cases when the compressor will not start when steam is first turned on, if steam is then turned off and allowed to remain off for one or two minutes, and then turned on quickly, it will start without the use of any oil, except that from the lubricator.

Compressor Groans.—Cause: (1) air cylinder needs oil. Remedy: (1) put some valve oil in air cylinder and saturate piston swab with valve oil, then replace it on the rod. Cause: (2) steam cylinder needs oil. Remedy: (2) increase lubricator feed.

Leakage past the air piston packing rings or past a discharge valve causes heating, destroys lubrication, and results in groaning. Piston rod packing dry and binding is another cause of groaning.

Uneven Strokes of the Compressor.—Cause: probably (1) leakage past air piston packing rings and sticky air valves; (2) unequal lift of air valves; (3) clogged discharge valve passages; or, (4) leaky air valves. Remedy: locate cause, if possible, and correct it by cleaning out clogged or dirty passages, adjusting lift of valves or replacing leaky valves or rings.

Slow in Compressing Air.—Cause: (1) leakage past the air piston packing rings, due to poor fit, or wear in cylinder or rings; (2) valves and passages dirty; or, (3) air suction strainer clogged. Remedy: (1) and (2). To determine which is causing the trouble

obtain about 90 lbs. air pressure, reduce the speed to from 40 to 60 single strokes per minute, then *listen* at the "Air Inlet" and note if air is drawn in during only a portion of each stroke, and if any blows back. If the latter, an inlet valve is leaking. If the suction does not continue until each stroke is nearly completed, then there is leakage past the air piston packing rings or back from the main reservoir past the air discharge valves. One of the latter leaking will cause an uneven stroke. Remedy: (3) clean strainer thoroughly.

Compressor Erratic in Action.—Cause: worn condition of valve motion. Remedy: replace it.

Compressor Heats.—Cause: (1) air passages are clogged; (2) leakage past air piston packing rings; or, (3) the discharge valves have insufficient lift. Remedy: (1) clean air passages; (2) renew air piston rings; (3) regulate lift of discharge valves to $\frac{3}{8}$ of an inch. A compressor in perfect condition will become excessively hot and is liable to be damaged if run very fast and continuously, for a long time.

Compressor Pounds.—Cause: (1) air piston is loose; (2) compressor not well secured, or causes some adjacent pipe to vibrate; (3) the reversing plate, 69, is loose; or, (4) the reversing rod or plate may be so worn that the motion of compressor is not reversed at the proper time. Remedy: repair and renew worn parts and tighten loose connections.

In connection with the problem of good maintenance for steam driven air compressors of this type, the heating of the air cylinder incident to air compression is perhaps the most important. The operation of the compressor continuously at high speeds or against excessive pressures inevitably results in high temperatures which tend to destroy the lubrication, causing the air cylinders to cut, and the groaning of the air compressor, besides filling the discharge passages with deposits from burnt oil, producing undesirable condensation of moisture in the system, and in general, reducing the over-all efficiency of the compressor.

Under normal conditions, the speed should not exceed 140 strokes per minute and such a speed should not be maintained continuously for any considerable time, as even this speed will eventually cause excessive heating.

It is therefore desirable, first, that the compressor be of ample capacity for the service desired; second, that it be well lubricated and otherwise maintained in good condition; and third, that leakage from any source whether within the air compressor itself or elsewhere in the system be minimized in every practicable way.

One of the most serious leaks is through the air cylinder stuffing box, as it not only greatly decreases the air delivered and, by the faster speed required, increases the heating, but it also causes pounding through loss of cushion. When tightening the packing, do not bind the rod, as to do so will damage both the packing and the rod. Be careful not to cross the gland nut threads.

Never remove or replace the upper steam cylinder head with the reversing valve rod in place, as to do so will almost invariably result in bending the rod. A bent rod is very liable to cause a "pump failure."

It is evident that a compressor cannot compress more air than it draws in, and not that much if there is any leakage to the atmosphere about the air cylinder. Bearing this in mind, practice frequently listening at the "Air Inlet" when the compressor is working slowly while being controlled by the governor, and wherever a poor suction is noted on either or both strokes locate and remedy the fault.

Any steam leakage that can reach the "Air Inlet" of the compressor should be promptly repaired, as this increases the danger of water entering the system.

Keeping the suction strainer clean is of the utmost importance, as even a slightly clogged strainer will greatly reduce the capacity where the speed is at all fast. A seriously or completely obstructed strainer, as by accumulated frost, aggravated by rising steam, will increase the compressor speed and will also be indicated by inability to raise or maintain the desired pressure.

It is an aid to good operation to thoroughly clean the air cylinder and its passages at least three or four times a year, by circulating through them a hot solution of lye or potash. This should always be followed by sufficient clean, hot water to thoroughly rinse out the cylinder and passages, after which a liberal supply of valve oil should be given the cylinder. Suitable tanks and connections for

performing this operation can easily be arranged in portable form. Never put kerosene oil in the air cylinder to clean it.

Ventilating Blowers.—Vessels are supplied with ventilating blowers for forcing in fresh air and drawing out foul air from the living spaces and various compartments. The blowers, now driven always by electric motors, are of the rotary fan type, revolving at a high rate of speed.

The ventilating fans are separate from those used for forced draught.

Main air ducts of light galvanized iron lead from the ventilating gear fore and aft, with branch ducts leading to the various rooms or compartments; the opening or louvres in the smaller ducts are covered with wire gauze. Automatic valves are fitted at bulkheads to prevent the flow of water from one compartment to another in case of accident.

Forced Draught Blowers.—The fans for forced draught are similar to the ventilating fans, but are arranged to draw from ventilating shafts and to discharge into a closed fire room, or closed ash pits, as the case may be. Practically all such blowers are now driven by direct-connected, high-speed electric motors, in large ships, and by turbines in all destroyers.

Refrigerating Machinery.—There are various systems of refrigeration, depending upon two general methods; one consisting of a change of state, physical, chemical, or both, and the other, the compression, cooling and expansion of an elastic gas, such as air.

The cold air system is based on the second principle and cold air machines are fitted on nearly all the cruisers and battleships in the navy. Carbonic acid machines are used on a few ships.

The working principle of air ice machines is based on the rise of temperature during the compression of air, and lowering of temperature about an equal amount during the expansion of the same. The machines, therefore, consist of an air compressor, which compresses the air and passes it into a cooling coil of pipe surrounded by circulated water. This removes the heat of compression and passes the compressed cooled air to the expander, a regular cut-off steam

engine into which the air is admitted during a portion of the stroke of the piston; the admission is then cut off and the air in the cylinder is expanded as the piston proceeds to finish its stroke.

During this expansion the air is cooled to a very low temperature, and the return stroke of the piston pushes it out, and pipes convey it to the place which is to be cooled by it. It was formerly usual to take air from the atmosphere and to refrigerate it without much compression. A difficulty was experienced in the lightness of the air, and its low capacity for cooling, which was lost very quickly. The machine was large and cumbrous and expensive in steam, and had to be placed immediately adjacent to the cold room.

A special set of attendants was needed unless the cold room was placed adjoining the engine room. The American air ice machine (Allen Dense Air Ice Machine) was constructed to overcome these difficulties, and is the type in general use in the navy.

Instead of taking air from the atmosphere or from a cold room, and after refrigeration discharging it again into the room, the Allen machine keeps a charge of air of five atmospheres pressure (sixty pounds gage pressure) contained in the machine and the conveying and refrigerating pipes, and uses the same over and over again, compressing it, cooling it in a copper coil surrounded by circulated water, expanding it in the expanding engine, and forcing it, when cold, through the conveying pipes to the cold room, where it also remains inside of pipes and does its refrigerating through the surface of the pipes. Then it passes back to repeat the performance.

Air of five atmospheres pressure has five times the capacity for absorbing heat as an equal volume of air of atmospheric pressure, and as it is conveyed in quite small pipes, the loss from radiation is small, and this makes it possible to have the machine in the engine room under the care of the engineers while they are attending to their regular duty, and to lead the cold air to any place which may be most advantageous, as the cold storage room, and scuttle butt. Referring to Fig. 204, and Plate 37, the machine consists of the following parts:

4. The Steam Cylinder, which furnishes the power required by the whole apparatus. To its crank shaft the air-compressor and expander are linked.

B. The Air Compressor Cylinder, which compresses the air to about three times the entering pressure, which compression causes the air to heat considerably. The cylinder is, therefore, surrounded by a water jacket, in order to keep the piston packings from withering up.

C. A Copper Coil in a Bath of Water. The compressed hot air, passing through the coil, cools to the temperature of the cooling water.

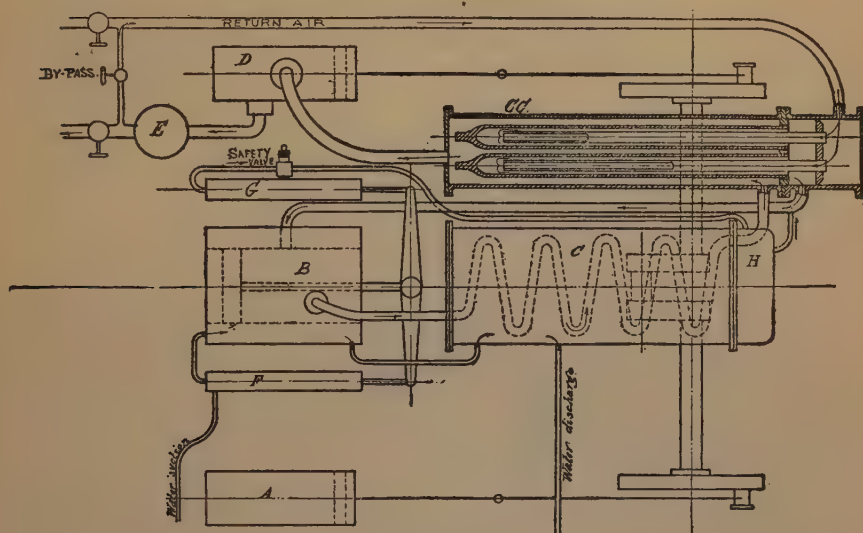


FIG. 204.

CC. The Return Air Cooler further cools this compressed air by means of the cold air returning from the cold storage room.

D. The Expander Cylinder, to which the cooled compressed air is admitted till it fills one-third of the volume of the cylinder. It is then cut off, and, the piston continuing its passage to the end of the cylinder, the air is expanded to one-third the compressed tension (*viz.*, to the same tension which it has when it enters the compressor). This expansion cools the air about as much as the compression heated it; therefore, the air leaves this cylinder at a very low temperature. This air is then discharged into a well-insulated pipe, which conveys it to the point where the power to absorb heat

is to be utilized. There the pipe service is exposed and heat is taken up from the air, or brine, as the case may be; after this utilization the air is returned to the machine, continually contained in pipes, and it enters the compressor for a new turn of compression, cooling and expansion. Before the expanded, refrigerated air goes into the conveying pipe after leaving the expander, it passes through a trap.

E is a trap which gathers the lubricating oil used in the cylinder, and also a slight amount of snow. The trap is provided with a heating pipe, and once in twelve or twenty-four hours this trap should be warmed up and the deposits be drawn off by the bottom cock. The machine is arranged, at the same time, to thaw out and blow into the trap any possible frozen deposits from the expander cylinder.

F is a common plunger water pump, which supplies the cooling water for trap *H*, for the bath around the copper coil *C*, and for the water jacket of the compressing cylinder *B*; the machine must be stopped if the water supply should fail.

G is a small supplementary air pump, which, at starting, primes the machine and the run of pipes with air to the requisite pressure, and which replaces losses of air from leaks through stuffing boxes and joints.

H is a small trap, which extracts moisture from this newly supplied air, so as to have it enter the machine as dry as possible. The gathered water must be drawn off occasionally from the bottom pet-cocks.

The machine is constructed to use an air pressure of 60 pounds in the conveying and refrigerating pipes, and the air compressor compresses this to 210 pounds. If these pressures cannot be maintained it is an indication that leaks have occurred, and the piston packings, and possibly the run of pipes, have to be examined. If the trap or any portion of the run of pipes is allowed to be choked by frozen oil or snow, or by closing of valves, the relation of the pressures will be disturbed. The cylinders have to be lubricated by the best quality of mineral lubricating oil.

The air pistons are packed with leather soaked in castor oil.

The air stuffing boxes contain first a few rings of Katzenstein soft metal packing rings, then a hollow oiling ring, then outer layers of fibrous packing, usually square Garlock packing. The oiling ring

is kept full of oil by a sight feed pressure lubricator which is connected by a pipe to the stuffing box.

The air forced out by the expander is commonly practically of about 35 to 55 degrees temperature below zero Fahrenheit, depending upon the temperature of the cooling water and upon internal leaks and frictions. The pipes lead it first through oil trap *E*, for purification, then to the ice-making box *I*, which consists of a casting, forming pockets, *T*, for the reception of sheet iron ice cans. This casting is set in a strong and tight jacket casting with internal bulkheads, formed so that the cold air which is led into the space between jacket and ice-can pockets must pass closely along the surfaces of the pockets.

The small space between the sheet iron ice cans and the inside of the pockets is filled with a solution of about equal weights of chloride of calcium and water, which withstands the cold without freezing. It provides a good conductor for the heat and keeps the cans from freezing fast in the pockets.

For larger apparatus, a wrought iron tank, filled with refrigerating pipes and ice cans, all immersed in the above brine, are used.

From the ice-making box the cold air is led to the cold storage chamber *K*, where it is passed through a system of refrigerating pipes *L*.

Frozen meat can be kept practically without change for an almost indefinite time. When kept at nearly the freezing point without change it may be kept for a number of weeks in good condition. A good practical rule for the amount of refrigerating pipes required in the meat chamber to keep it at the freezing point is: One square foot of pipe surface for every $2\frac{1}{4}$ to $2\frac{3}{4}$ square feet of interior surface of well-insulated meat chamber, omitting interior divisions. It is necessary to arrange the piping so that the air in them is compelled to pass all surfaces with fair velocity.

From the cold storage the cold air goes to the refrigerating pipes in the drinking-water butt *M*, passing first to the bottom layer and then gradually upward.

Directions for Operating.—On starting the machine, have the blow valves of the expander and the pet-cocks of the various traps open until no more grease or water discharges.

The two $1\frac{1}{2}$ or 2-inch valves of the main pipes must be open and the 1-inch by-pass pipe closed; also the $\frac{1}{2}$ -inch hot-air valves from the compressor to the expander cylinder must be closed.

Be sure that the circulating water is in motion.

The full pressure is 60-70 lbs. low pressure, and 210-240 lbs. high pressure.

During the running, open the pet-cocks of the water-trap, which takes the water out of the air from the primer pump, frequently enough so that it will never be more than half filled. If the water should be allowed to enter the main pipes, it is liable to freeze and clog at the valves.

By keeping all stuffing boxes well lubricated by the lubricator cups, the pressures are easily maintained with but little screwing up of the packing.

If the low-air pressure is not maintained, the fault is almost always due to leaks at the stuffing boxes. Under all circumstances it is due to some leak into the atmosphere, as the primer pump valves have been rarely found at fault.

The pistons of the compressor and expander-cylinders are packed with cup leathers, which commonly last about one or two months of steady work. When these leathers give out, the high pressure decreases in relation to the low pressure, and the apparatus shows a loss of efficiency. A leak at any other point of high pressure into low pressure will have the same effect.

These packing leathers are made of white-oak-tanned leather of somewhat less than $\frac{1}{8}$ -inch thickness. They are cut $\frac{5}{8}$ -inch larger in diameter than the cylinders. The leathers must be kept soaked with castor oil and must be well soaked in that before using; and a tin box containing spare leathers and castor oil must be kept on hand.

Once, or sometimes twice a day, it is necessary to clean the machine by heating it up and blowing out all the oil and ice deposits. This is done as follows:

The 1-inch valve of the by-pass is opened. Then the two $1\frac{1}{2}$ or 2-inch valves in the main pipe are closed; then the two $\frac{1}{2}$ -inch valves in the hot-air pipe from the compressor chest to the expander are opened, and the $1\frac{1}{4}$ -inch valve of expander inlet is closed partly;

then the live steam is let into the jacket of the oil-trap slowly, keeping the outlet from the steam jacket open enough to drain the condensed steam.

Run in this manner for about one-half hour; during this time frequently blow out the bottom valve of the oil-trap, also the blow-off from the expander, until everything appears clean. Then shut off the steam and drain connections of the jacket of the trap and the hot-air pipe from the compressor to the expander. Then open the two $1\frac{1}{2}$ -inch valves in main pipes. Then close the 1-inch by-pass pipe and all pet-cocks and run as usual.

Whenever opportunity offers to blow out the manifolds of the cold storage and the ice-making box (that is, whenever they are thawed), this should be done.

Carbonic Acid Refrigerating Apparatus.—This machine, Fig. 205, has a greater economic efficiency than the dense air machine and is fitted in many ships of the mercantile marine, but owing to the difficulty that naval vessels may experience in the replenishment of the carbonic anhydride, it has not been much used in our ships.

Carbonic anhydride under atmospheric pressure has a boiling point of 120 degrees below zero, and with high pressures the boiling point rises. On passing into a state of vapor, heat is taken from the liquid itself and the surroundings, thus lowering the temperature.

In the utilization of these qualities in refrigeration, a carbonic acid refrigerating machine consists of a compressor, a condenser and an evaporator. The compressor takes the expanded gas from the evaporator, compresses it and forces it into a condenser composed of pipes surrounded by circulating sea water. Here the compressed gas is cooled and liquefied and passed into the evaporator coils which are surrounded by brine. In vaporizing in the evaporator coils, heat is absorbed which is taken from the surrounding brine. This brine after being cooled down by this process is forced by a brine pump through coils in the refrigerating room. Any moisture in the air condenses on these pipes, forming thick coatings of frost. A large reserve of cooling medium due to the cold brine retained in the brine pipes enables the refrigerating room to be kept cold

for a considerable time, should it be found necessary to stop the machine. A small amount of carbonic anhydride must be added from time to time to make good the loss from leakage.

Engine Room Telegraphs.—The orders in regard to working the engines are transmitted from deck through connections to a mechanical repeating telegraph fitted for each engine room. The dials are located in a convenient place at the working platform and connected

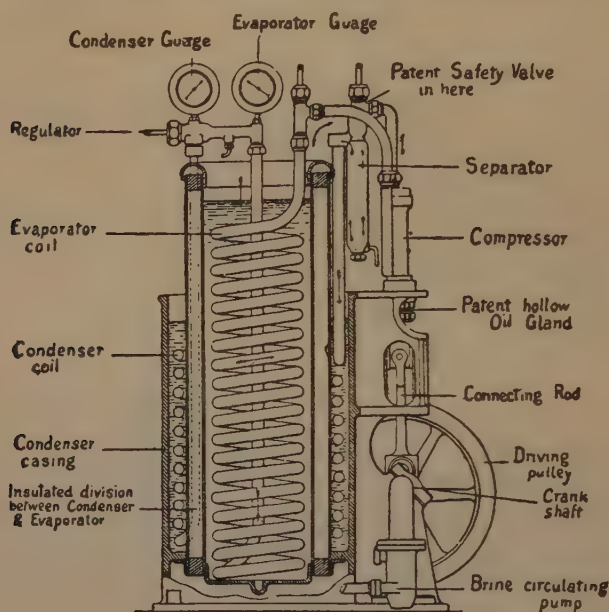


FIG. 205.

to transmitters on the bridge and in the conning tower and pilot house. A bell is fitted in the receiver to call attention to the moving of the telegraph and a tell-tale on the transmitter indicates that the signal has been received and that the engine is acting in accordance with the signal.

In the installation of the receivers in the engine room and the transmitters on deck, particular care should be taken in running the leads of the wire or chains, so that the chance of derangement shall be minimized.

Speaking tubes with mouthpieces are fitted between engine and fire room and between the deck and the engine room to facilitate communication. These tubes are fitted with electrical call bells and indicators or whistles in order to attract attention.

A large gong is also fitted for each engine room and mechanically connected with bell pulls on the upper deck and bridges, for use in case of derangement of the engine room telegraph.

CHAPTER XV.

ENGINE BALANCING.

Preliminary.—The relation of machinery in motion to the vibration of ships has only recently been placed on a scientific basis. A great increase of power and speed has practically forced a study of the subject with a view to reducing the serious effects of unbalanced forces generated by propelling engines. The rotative speeds of engines built a few decades ago were not great enough to develop large forces, and besides the types were not favorable to accurate balance. Hence, little attention was paid to balancing reciprocating or rotating masses, even though the vibration set up by them seemed at times extremely disagreeable. The higher speeds demanded for naval vessels have been obtained in part by an increase of rotative speed. In fact, this has been one of the principal factors in reducing the weight of machinery, and thus, in obtaining large powers within a given limit of displacement. A useful illustration of the growth of marine engines is afforded by the first really high-speed steamer ever built. During the Civil War, the Navy Department designed and built the *Wampanoag* for the purpose of overtaking and capturing Confederate commerce destroyers. This ship averaged on her trial trip of 36 hours about $17\frac{1}{2}$ knots an hour, but she was, nevertheless, a practical failure. The hull was of wood, and could not withstand the unbalanced forces of high rotative speeds. Her engine was, therefore, designed for thirty revolutions per minute geared by very heavy spur wheels to a propeller shaft making seventy-five revolutions. The machinery was thus enormously heavy, and the coal supply was limited. If the engine had been connected directly to the shaft and run at seventy-five revolutions per minute, the power could have been increased two and a half times without material change of weight, but the unbalanced forces would have been increased six and a quarter times. The vibration set up in the wooden hull would probably have shaken the ship to pieces. While a high-speed engine might have been designed to answer all purposes, it is

doubtful if engineering practice could have made it successful until within a few years. The rotative speed has increased four and five fold for the larger vessels since the Wampanoag was built, and it has gone up to five and six hundred revolutions per minute for torpedo boats and other craft requiring great power in small compass. A knowledge of balancing engines has therefore become an important part of the marine engineer's equipment.

Two Sets of Moving Parts.—In the ordinary steam engine there are two kinds of moving masses, those which rotate, and those which move in a straight line. The former are usually treated as if their rotation took place at a uniform rate throughout a revolution. This supposition is not strictly correct, yet an engine which is balanced for uniform rotative speed is also balanced for variable speed, and it is therefore unnecessary to introduce quantities which would serve only to complicate the subject.

The reciprocating parts come to rest at the end of every stroke, having their maximum velocity of translation in the neighborhood of mid-stroke. The connecting rod partakes of both motions; the wrist pin end moving parallel to the guides, and the crank pin end in a circle. It may without sensible error be divided into two parts for the purpose of this investigation. The simplest division is to place one-half the weight of the rod on each pin. A more accurate distribution is to allot the respective weights just as if the rod were supported in a horizontal position by the two pins. The weight taken by each pin will then be inversely proportional to its distance from the center of gravity of the rod. As marine connecting rods are generally of one type, this amounts to adding six-tenths of the weight of the rod to the crank pin as a revolving mass, and four-tenths the weight to the wrist pin as a reciprocating mass. A more accurate analysis of the motion of the connecting rod may be found in Professor Dalby's book on *Balancing Engines*.

Variations in Crankshaft Moments and Torsional Vibration.—Recalling the chapter on Crank Effort Diagrams, the forces acting upon the framing of an engine may be divided into two varieties, those due to steam pressure, and those due to the weight of the moving masses; the latter again being subdivided, as stated above,

into rotating and reciprocating parts. The frictional resistances are neglected here, as they do not in any way affect the balancing problem. As the purpose of this chapter is to show the effects of steam upon the hull of a ship, and how to neutralize them, the forces, including steam pressure, may with profit be taken up in the order set down.

Let Fig. 220 represent one unit of a vertical marine engine;

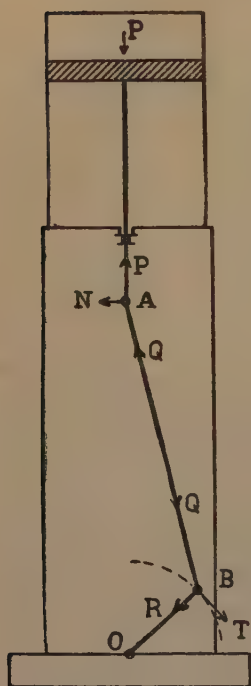


FIG. 220.

P , the pressure transmitted through the piston rod to the crosshead; Q , the pressure along the connecting rod, and consequently the pressure upon the crank pin. At the crank pin, B , the pressure Q may be decomposed into a tangential pressure, T , and a radial pressure, R . The turning moment on the shaft is then $T \times OB$, and this is transmitted through the line shaft to the propeller, where it is exerted upon the water. The pressure Q has an equal reaction upon the wrist pin where it may be decomposed into a force parallel to the slides and equal to P , and a force, N , normal to the slides. The four forces T , R , P and N , thus hold one another in equilibrium. If moments be taken about O , R and P disappear, and $T \times OB = N \times OA$; from which it is plain that the reaction N exerts upon the engine framing a capsizing moment equal to the turning moment on the crank shaft, but in the opposite direction. As the engine bed plate is bolted to the ship, this capsizing moment expends itself on the hull. Further-

more, as shown in Chap. XI, this moment is variable, and passes through zero and a maximum twice during one revolution. Even the combined moments of several units show great variation. Since this variation is periodic, its effects may become very disagreeable, in case the periods synchronize with the natural periods of vibration of the hull, or of the engine framing itself. On the trial trip

of one of the cruisers where, by faulty valve setting, a large fraction of the work was thrown on one cylinder, the turning moment was very irregular and the vibrations were so violent that the searchlights had to be lashed in place.

In general, the capsizing couple on the engine may be considered in three aspects:

First. It tends to heel the ship as a whole in the reverse direction to the propeller's motion. If the periodic variation of the turning moment synchronizes with the natural period of rolling, the ship will tend to increase her angle of oscillation. This tendency may be lessened by the use of twin screws with shafts revolving in opposite directions, although it still exists when the maximum turning moment of one engine occurs at the same time as the minimum turning moment of the other.

Second. The periodic impulses tend to twist the hull locally. Each impulse upon the engine keelsons must be transmitted through the ship whose parts resist the motion by reason of their mass. There is thus a torsional vibration set up which may appear in the forward part of the ship, while the midship section is comparatively undisturbed. Again, the vibration may become violent if the periods synchronize.

Third. The effect of the variable moment may appear in the engine or shaft. Every part has its own period of oscillation and the general effect, unless the engine framing is well braced to take the capsizing moment in each cylinder, may be serious. An example of how badly an engine may behave was supplied by the early trial trips of the *Dolphin*, where the low-pressure cylinder jarred the indicators out of their places, and it was extremely difficult to take an indicator card.

The remedy in all cases lies in well-designed framing, and in a uniform turning moment on the shaft. In changing crank angles to secure a better balance of reciprocating parts, the latter fact must not be forgotten. The practice of the Navy Department has in general given satisfactory results. There are four cylinders on each crank shaft and the cranks are placed 90 degrees apart. With the horse power well distributed in the cylinders, and a good sequence of cranks, the moment on the crank shaft should not vary more than 15 per cent on each side of the average, and the torsional vibration should be small.

Origin of Unbalanced Forces.—The forces due to the steam pressure are always balanced excepting so far as they vary in their effects upon the turning moment. Steam pressure in the cylinder is exerted in every direction alike, and the pressure transmitted to the shaft and bed plate is counteracted by the cylinder and framing.

The forces due to the moving masses are, however, in the nature of outside or free forces independent of the steam pressure and of the horse power developed. Although the reciprocating parts have been discussed in a previous chapter, something more remains to be said in reference to their effect upon the engine.

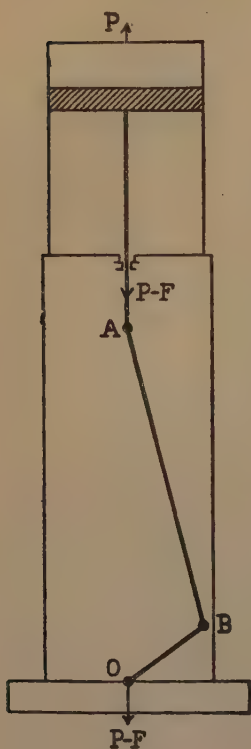


FIG. 221.

In Fig. 221, let P be the steam pressure on the piston and the equal opposite reaction on the cylinder head, and F the pressure required to produce the acceleration of the reciprocating parts. Then, $P-F$ is the pressure transmitted through the piston rod to the crank shaft. Since the pressure on the cylinder heads thus exceeds the pressure on the engine bed plate by the amount F , there will be an unbalanced force equal to F tending to lift the entire engine off its support, or at the other end of the stroke, to drive it into the support. A parallel to this is the recoil of a gun when a projectile is driven out of it by the pressure of the powder. The only difference is that the acceleration of the piston is determined by its connection with the crank pin, while the acceleration of the

shell is controlled only by the excess of the powder pressure over the resistance in the bore.

There is also an unbalanced pressure on the wrist pin and crank pin at right angles to the plane of the slides, springing from the angular acceleration of the connecting rod. Its tendency to capsize the engine is small in comparison with the turning moment on the

crank shaft, and it is usually neglected. The value of F , however, is large, and must be considered in the design of an engine. It is always exerted in a plane parallel to the center line of the cylinder, and is like a blow struck twice in a revolution, giving rise, in many cases, to violent oscillation of the engine foundation.

The rotating masses differ from the reciprocating masses in that the forces generated by them turn with the shaft. From the principle of mechanics, it is known that a rotating body exerts a radial pull or centrifugal force away from the axis of rotation, and that the pull is the same as if the entire body were concentrated at its center of gravity. Letting N be the number of revolutions per minute; r , the distance of the center of gravity from the axis of rotation; W , the weight of the body, and g the acceleration due to gravity, the centrifugal force G will be given by the formula:

$$G = \frac{W}{g} \times \frac{\pi^2 N^2}{900} \times r.$$

This constitutes an unbalanced force tending to move the shaft out of its supports. When the center of gravity of the combined rotating masses is above or below the axis of the shaft, the tendency is to lift or depress the hull of the ship and thus to set up vertical vibration; when it is to the right or left of the shaft axis, the forces act laterally. As the shaft revolves the forces revolve, and consequently have both horizontal and vertical components at all times.

Inasmuch as forces due to the moving masses, reciprocating and rotating, are periodic in their nature, their effect upon the hull of a ship depends upon their falling into synchronism with the natural periods of vertical or horizontal oscillation. Hence a ship may vibrate seriously at some speeds and very little at others. The only method of preventing undesirable vibration due to the machinery is by balancing completely all moving masses. To some extent this has been accomplished in the modern marine engine, but much remains to carry out practically the theories which are now well understood.

Rotating Masses.—The rotating parts are the simplest to deal with, and are here presented first. In what follows, the technical terms and notation are those in common use, but a few require special definition.

A *radial plane* is one at right angles to the axis of rotation. An *axial plane* is one containing the axis. A *plane of reference*, either radial or axial, is one from which the measurements are taken. The angle θ will be the angle through which the body has rotated from some axial reference plane.

The problem which occurs in practice may be stated generally as follows: Given the position and weight of any number of cranks, crank pins and eccentrics on a rotating shaft, to find the amount and position of weights required to counterbalance them. To have complete equilibrium, there must be no resultant force, and no resultant couple. The work may be simplified by reducing all the masses to equivalent masses, having their centers of gravity the same distance from the axis, usually the length of the crank, r .

Let W_1 be the weight of a crank web or any other part, and r_1 the distance of its center of gravity from the shaft axis. Then W , the equivalent weight at the center of the crank pin will be found from the equation,

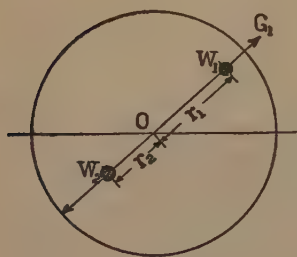


FIG. 222.

$$\frac{W\pi^2 N^2}{900g} r = \frac{W_1\pi^2 N^2}{900g} r_1$$

$$Wr = W_1 r_1 \therefore W = \frac{W_1 r_1}{r}$$

The mass W may be treated in every way as the equivalent of W_1 , and it may replace it in the problem.

There are three methods of balancing the set of rotating masses: First, by placing a sufficient counterbalance in the radial plane of each mass; second, by adding two revolving weights outside of the engine framing at the proper angle for counterbalancing all the given parts; third, by adjustment of the relative positions and angles of masses so that they counterbalance one another.

First Solution for Rotating Masses.—Let W_1 , Fig. 222, be the weight of a crank pin, the adjoining crank webs and their share of the connecting rod reduced to the crank pin center. The centrifugal force is then

$$G_1 = \frac{W_1\pi^2 N^2}{900g} r_1$$

This may be completely counterbalanced by another weight W_2 whose center of gravity is placed in the same radial plane as W_1 , and directly opposite to it at a distance r_2 such that

$$\frac{W_2 \pi^2 N^2}{900 g} r_2 = \frac{W_1 \pi^2 N^2}{900 g} r_1.$$

Then $W_2 r_2 = W_1 r_1$

and $W_2 = \frac{W_1 r_1}{r_2}$, or $r_2 = \frac{W_1 r_1}{W_2}$.

Either W_2 , or r_2 , may be assumed.

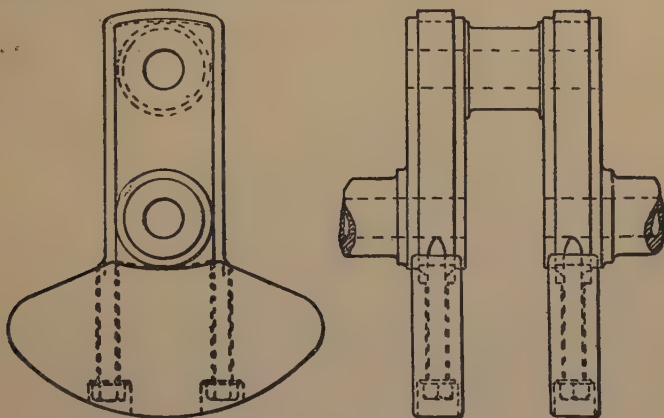


FIG. 223.

The weight is usually divided equally between the two crank webs and bolted on as shown in Fig. 223.

The weight W_1 may also be counterbalanced by two or more weights, properly distributed on the crank disc, as in the case of a locomotive driving wheel. The amount or position of, say two weights W_2 and W_3 , may be determined exactly as three forces meeting in a point are dealt with in Statics. In Fig. 224 let θ be the angle between the axial plane through the center of gravity of W_1 , and the initial axial plane OX , α_2 and α_3 , the angles between the axial planes through

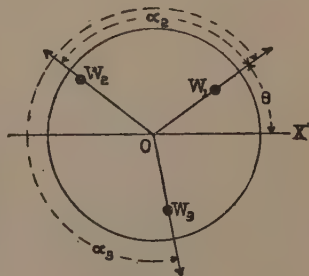


FIG. 224.

the centers of gravity of W_1 , W_2 and W_3 , respectively, and r_1 , r_2 , r_3 , the respective distances of the centers of gravity from the axis. Decompose the centrifugal forces into components parallel and at right angles to OX , and place the algebraic sum equal to zero. Then

$$\frac{\pi^2 N^2}{900 g} (W_1 r_1 \cos \theta + W_2 r_2 \cos(\theta + a_2) + W_3 r_3 \cos(\theta + a_3)) = 0$$

$$\frac{\pi^2 N^2}{900 g} (W_1 r_1 \sin \theta + W_2 r_2 \sin(\theta + a_2) + W_3 r_3 \sin(\theta + a_3)) = 0$$

Dividing through by the constant $\frac{\pi^2 N^2}{900 g}$, and making $\theta = 0$, since the equations hold good for any crank angle, we get

$$W_1 r_1 + W_2 r_2 \cos a_2 + W_3 r_3 \cos a_3 = 0$$

$$W_2 r_2 \sin a_2 + W_3 r_3 \sin a_3 = 0$$

In these equations only two quantities may be unknown; the others must be assumed or given. The angles a_2 and a_3 are usually known, leaving the values of $W_2 r_2$ and $W_3 r_3$ to be found. This

problem may easily be solved graphically by the triangle of forces, simply by treating $W_1 r_1$, $W_2 r_2$, $W_3 r_3$, as forces, and laying the first out to scale.

The weight W_1 cannot be balanced by one or more weights in a single radial plane unless the plane be that containing W_1 . Suppose, in Fig. 225 a single weight,

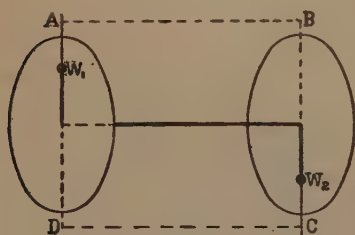


FIG. 225.

W_2 , be placed exactly opposite to W_1 , but in a different radial plane, r_1 and r_2 , being the respective distances of the centers of gravity from the axis, and a the distance apart of the radial planes. Also, let

$W_1 r_1 = W_2 r_2$. Then the centrifugal forces $\frac{W_1 \pi^2 N^2}{900 g} r_1$ and

$\frac{W_2 \pi^2 N^2}{900 g} r_2$ constitute a centrifugal couple whose moment is

$\frac{W_1 \pi^2 N^2}{900 g} r_1 a$, tending to turn the shaft end for end in the axial

plane $ABCD$. This is the condition of many engines with cranks placed 180° apart. In counterbalancing it is plain that couples as well as simple forces must be neutralized.

The weight W_1 may be balanced by two or more weights not in the same radial planes providing that no axial couple remains. If only two are used, they must lie in the axial plane containing W_1 . This case is solved exactly as a lever or beam with three parallel forces applied to it.

Let O_1, O_2, O_3 , Fig. 226, be the three radial planes containing the weights W_1, W_2 and W_3 , whose centers of gravity are respectively r_1, r_2 and r_3 from the axis. Take O_2 as the reference plane, and let a_2, a_3 be the respective distances of O_1 and O_3 from it. Omitting the constant factor $\frac{\pi^2 N^2}{900g}$ and taking moments about O_2 , the equation $W_1 r_1 a_2 = W_3 r_3 a_3$ determines the value of any one quantity in $W_3 r_3 a_3$. Also, $W_2 r_2 + W_3 r_3 = W_1 r_1$, from which $W_2 r_2$ may be

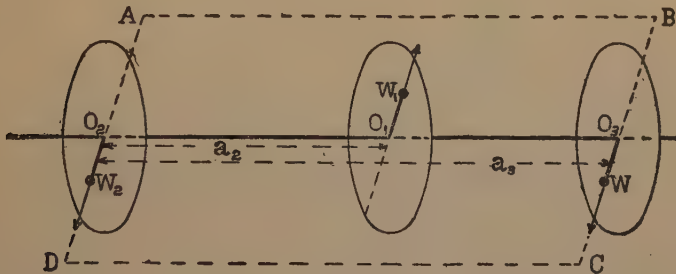


FIG. 226.

round. Usually the values of a_2 and a_3 are given. This case represents three cranks set 180° apart and perfectly balanced; or it represents one crank W_1 , with two counterbalance weights outside of the engine framing.

Second Solution for Rotating Masses.—The last example in the previous paragraph is a special case of the general method of balancing any number of rotating masses by two weights added to the crank shaft. These weights are placed where they interfere least with other parts, usually in discs at each end of the crank shaft.

Using the same notation as heretofore, let W_1, W_2, W_3 , etc., Fig. 227, be a series of weights in the radial planes O_1, O_2, O_3 , etc., with centers of gravity r_1, r_2, r_3 , etc., from the axis. Let the axial

planes through the centers of gravity make angles $\alpha_1, \alpha_2, \alpha_3$ etc., with the reference plane $ABCD$, and let a_1, a_2, a_3 , etc., be the distances of the radial planes from a reference plane O_n . The axial plane through the center line of the cylinders is usually taken as a reference plane for angles.

The problem resolves itself into finding two weights W_n and W_m placed respectively in the reference plane O_n , and a second plane to the right whose distance a_m from the reference plane is given. Let θ_n and θ_m be the angles between the axial plane $ABCD$ and the planes of the two unknown weights. Then for complete equilibrium there are four equations derived from the components of centrifugal forces taken parallel and at right angles to $ABCD$.

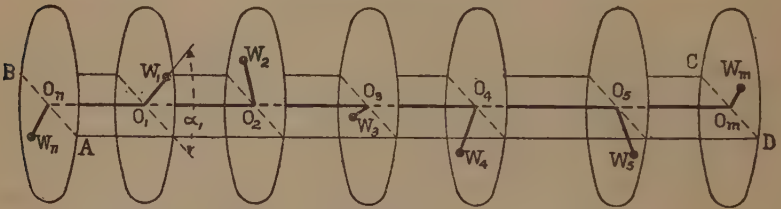


FIG. 227.

The first two are the algebraic sum of the simple components, and the second two the algebraic sum of the moments about O_n . Omitting the constant factor $\frac{\pi^2 N^2}{900 g}$, the equations are:

$$W_n r_n \cos \theta_n + W_1 r_1 \cos \alpha_1 + W_2 r_2 \cos \alpha_2 + \text{etc.} + W_m r_m \cos \theta_m = 0$$

$$W_n r_n \sin \theta_n + W_1 r_1 \sin \alpha_1 + W_2 r_2 \sin \alpha_2 + \text{etc.} + W_m r_m \sin \theta_m = 0$$

$$W_1 r_1 a_1 \cos \alpha_1 + W_2 r_2 a_2 \cos \alpha_2 + \text{etc.} + W_m r_m a_m \cos \theta_m = 0$$

$$W_1 r_1 a_1 \sin \alpha_1 + W_2 r_2 a_2 \sin \alpha_2 + \text{etc.} + W_m r_m a_m \sin \theta_m = 0$$

If equivalent weights at the crank pin be substituted for the actual weights, the equations become:

$$W_n \cos \theta_n + W_m \cos \theta_m = -\Sigma W \cos \alpha$$

$$W_n \sin \theta_n + W_m \sin \theta_m = -\Sigma W \sin \alpha$$

$$W_n a_n \cos \theta_n + W_m a_m \cos \theta_m = -\Sigma W a \cos \alpha$$

$$W_n a_n \sin \theta_n + W_m a_m \sin \theta_m = -\Sigma W a \sin \alpha$$

There are thus four unknown quantities, W_n , W_m , θ_n and θ_m , to be found. From the last two

$$\tan \theta_m = \frac{\Sigma W a \sin \alpha}{\Sigma W a \cos \alpha}$$

and $W_m a_m = \{(\Sigma W a \cos \alpha)^2 + (\Sigma W a \sin \alpha)^2\}^{\frac{1}{2}}$.

Substituting these in the first two, and W_n and θ_n are determined by similar equations. This solution is perfectly general, and may be applied to any system whatever. Care should be exercised in selecting the two reference planes O_n and O_m .

One example will suffice to make the application clear. Take the ordinary triple-expansion engine, Fig. 228, with all main rotating weights the same, and cranks placed 120° apart. Then $\alpha_1 = 0$, $\alpha_2 = 120^\circ$, $\alpha_3 = 240^\circ$ and $W_1 = W_2 = W_3 = W$.

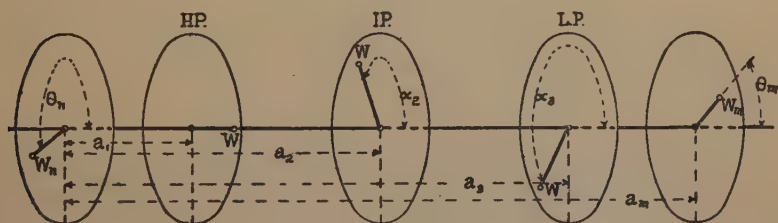


FIG. 228.

Substituting in the four fundamental equations we get

$$W_n \cos \theta_n + W_m \cos \theta_m = 0$$

$$W_n \sin \theta_n + W_m \sin \theta_m = 0$$

$$W_m a_m \cos \theta_m = -\frac{W}{2} (2a_1 - a_2 - a_3)$$

$$W_m a_m \sin \theta_m = -\frac{W}{2} \sqrt{3} (a_2 - a_3)$$

Supposing that $a_1 = 4'$, $a_2 = 6'$, $a_3 = 9'$ and $a_m = 12'$, the last two equations become

$$W_m a_m \cos \theta_m = +\frac{7}{2} W$$

$$W_m a_m \sin \theta_m = \frac{3\sqrt{3}}{2} W$$

and

$$\tan \theta_m = \frac{3\sqrt{3}}{7} = .7423$$

$$\theta_m = 36^\circ 35'$$

$$W_m = W \times .363$$

From the first two equations, then,

$$W_n \cos \theta_n = -\frac{7}{24} W$$

$$W_n \sin \theta_n = \frac{3\sqrt{3}}{24} W$$

and $\theta_n = 216^\circ 35'$, $W_n = W \times .363$.

The first two equations being equal to zero showed at once that the simple forces were balanced and that only a couple remained. This would be counterbalanced by the couple $W_m a_m = 4.356 W$ foot-pounds placed in an axial plane making $36^\circ 35'$ with the reference plane.

It can also be shown in the same way that the four-cylinder engines with cranks 90° apart are balanced for simple centrifugal forces but require a centrifugal couple supplied by additional weights to balance them completely.

Third Solution for Rotating Masses.—If there are a sufficient number of cranks and revolving weight, an engine can be made self-balancing so far as the rotating masses are concerned without additional weights. To accomplish this may involve a sacrifice of the uniformity of turning moment on the shaft, unless the weights belonging to the cranks vary. Usually the dimensions of all cranks and pins are the same for the sake of interchangeability, and it is therefore not easily possible to make less than six cranks self-balancing without giving up something more valuable. The subject may easily be studied by means of the fundamental equations. If the engine is perfectly balanced there is no resultant force or couple, and

$$\Sigma W \cos \theta = 0$$

$$\Sigma W \sin \theta = 0$$

$$\Sigma W a \cos \theta = 0$$

$$\Sigma W a \sin \theta = 0$$

The reference planes should be so chosen as to pass through the center of one of the crank pins or weights.

It will be found in ordinary marine practice that simple centrifugal forces are carefully balanced as in the triple-expansion

three and four cylinder engines, but that little attention is paid to upending couples. The additional weights necessary are not readily given where displacement is so valuable for coal, guns and equipment. The couples are reduced to their smallest values by placing the cylinders as close together as possible.

Graphic Solution.—The graphic method hitherto mentioned, may be extended to the general problem especially where two engines already designed are to be compared. It is extremely simple. Suppose all the weights to be projected on the radial reference plane,

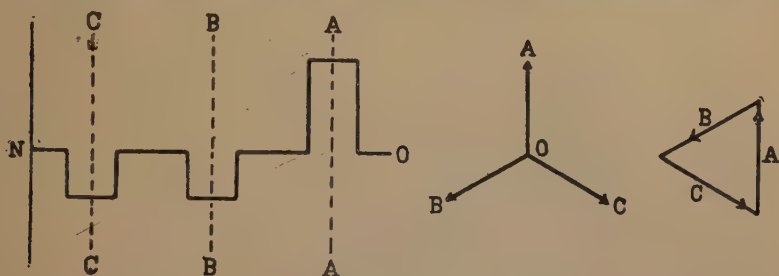


FIG. 229.

Force Diagram—Three Crank Engine. N. Reference plane for moments.

and lines to be drawn from the center of the shaft through their centers of gravity. Lay off upon these lines to any scale the equivalent weights W_1, W_2, W_3 , etc., reduced to the crank pins, and treat them as forces radiating from a point. The resultant found by the polygon of forces will represent the amount and direction of the single weight to be balanced, in order that the whole system of simple centrifugal forces may be balanced. In the same way lay off the moments W_1a_1, W_2a_2, W_3a_3 , etc., upon the respective radial lines drawn as before, and again treat these as central forces whose resultant

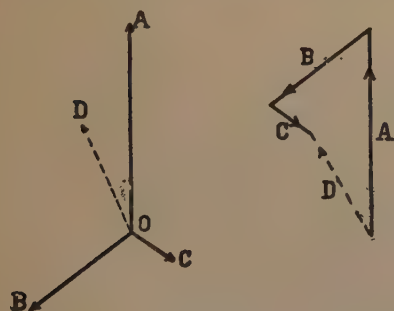


FIG. 230.

Moment diagram about N. D = resultant moment.

W_2a_2, W_3a_3 , etc., upon the respective radial lines drawn as before, and again treat these as central forces whose resultant

can be found by the polygon of forces. The resultant multiplied by the factor $\frac{\pi^2 N^2 r}{900 g}$ will be the unbalanced couple acting on the shaft. The three and four crank engines are illustrated in Figs. 229, 230, 231 and 232 without further explanation.

Reciprocating Masses.—The unbalanced forces due to the reciprocating weights may be investigated in the same way as the

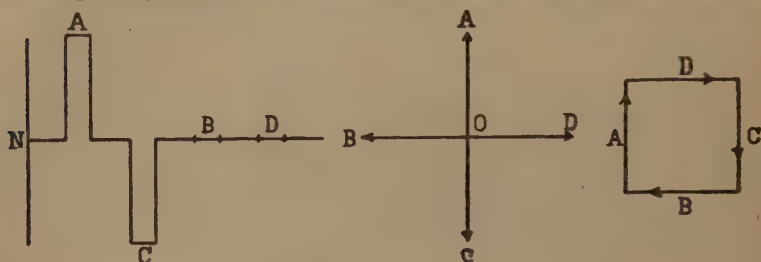


FIG. 231.

Force diagram—4-crank engine.

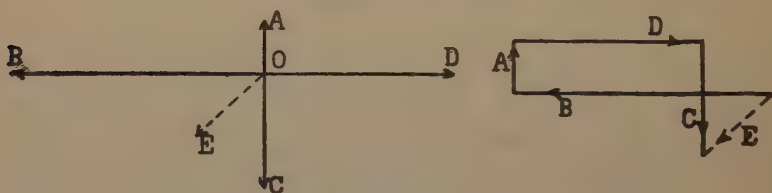


FIG. 232.

Moment diagram about N. E = resultant moment.

centrifugal forces. The expression for F , the force required to produce the acceleration of these weights, given in Chap. XI, is

$$F = \frac{W\pi^3 N^2 r}{900 g} \left\{ \cos \theta + \frac{\cos^2 \theta - \sin^2 \theta}{\sqrt{\left(\frac{l}{r}\right)^2 - \sin^2 \theta}} + \frac{\sin^2 \theta \cos^2 \theta}{\left(\left(\frac{l}{r}\right)^2 - \sin^2 \theta\right)^{\frac{3}{2}}} \right\}$$

in which $\frac{W\pi^3 N^2 r}{900 g}$ is the centrifugal force of the reciprocating parts considered as rotating masses placed at the center of the crank pin and $\frac{l}{r}$, the ratio of the connecting rod to the crank. The factor in

the brackets may be expanded into a series in terms of the cosines of the angle θ and its even multiples.

Thus, $F = G(\cos \theta + A \cos 2\theta + B \cos 4\theta + C \cos 6\theta + \text{etc.})$.

In this equation $G = \frac{W\pi^2 N^2 r}{900 g}$

$$A = \frac{r}{l} + \frac{1}{4} \frac{r^3}{l^3} + \frac{15}{128} \frac{r^5}{l^5} + \text{etc.}$$

$$B = -\frac{1}{4} \frac{r^3}{l^3} - \frac{3}{16} \frac{r^5}{l^5} - \text{etc.}$$

$$C = +\frac{9}{128} \frac{r^5}{l^5} + \text{etc.}$$

Substituting the ordinary value $\frac{r}{l} = \frac{1}{4}$, in these equations, it will be seen that the powers of $\frac{r}{l}$ higher than the first are negligible. Hence

$$F = G(\cos \theta + \frac{r}{l} \cos 2\theta)$$

The force required to produce the acceleration of the reciprocating masses is therefore composed of two parts, $G \cos \theta$, called primary, and $G \frac{r}{l} \cos 2\theta$, called secondary. The first $G \cos \theta$ is simply the component of the centrifugal force parallel to the center line of the cylinder, and the second $G \frac{r}{l} \cos 2\theta$ is the component of the centrifugal force $G \frac{r}{l}$ parallel to the same line but taken at twice the angle. The former passes through a numerical maximum twice during a revolution and the latter four times. The curves representing them may be plotted as primary and secondary force curves by laying off the angles along a horizontal axis, and the successive values of $G \cos \theta$ and $\frac{Gr}{l} \cos 2\theta$ vertically. They simply produce cosine or sine curves, the second being of double frequency.

In Fig. 233 *a-a-a* is the primary curve of the Missouri's H. P. unit, *b-b-b*, the secondary curve, and *c-c-c*, the combined curve representing $F = G(\cos \theta + \frac{r}{l} \cos 2\theta)$ for all crank angles throughout one revolution. Thus the acceleration, or force, curve is a

cosine curve modified by a smaller one of double frequency. The primary curve neglects $\frac{r}{l}$ and is therefore the force curve for a straight yoke engine, or one with the connecting rod infinitely long. The secondary curve expresses the modification due to the angularity of the connecting rod.

As stated in Chap. XI, $F = G\left(1 + \frac{r}{l}\right)$ when $\theta = 0$, $F = -G \times \left(1 - \frac{r}{l}\right)$, when $\theta = 180^\circ$ and $F = 0$ when $\theta = 77^\circ$ for a value of $\frac{r}{l} = \frac{1}{4}$.

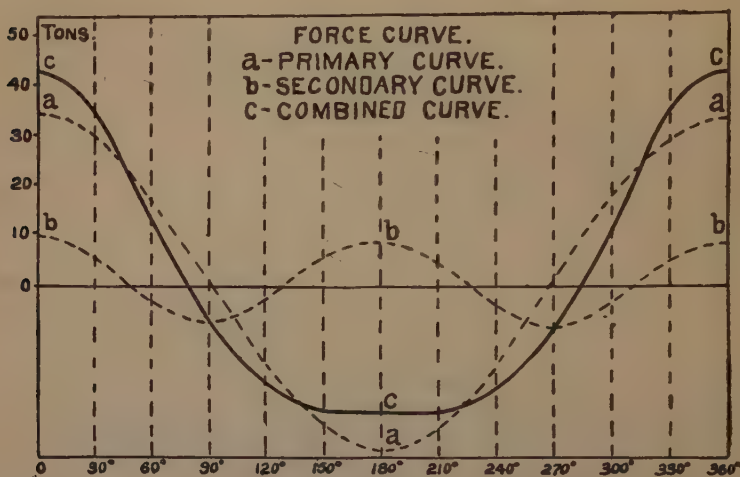


FIG. 233.

Thus it is plain that the accelerating force of the reciprocating parts cannot be balanced by a rotating mass. It can be balanced on either dead point, but since the centrifugal force is constant, new unbalanced forces are thereby introduced. The weights placed in the spokes of a locomotive driving wheel are used to balance the reciprocating parts at the expense of the road bed; for the vertical components of the unbalanced centrifugal forces thus developed tend alternately to lift the wheels and to pound the rails.

If there are several cylinders with cranks set at different angles, the reciprocating weights balance one another in part. The total

accelerating force for the engine may be found simply by adding together the free forces of the different units. This is best done by drawing the accelerating force curves at their proper angles, and combining them into one curve showing for every angle the unbalanced force. Fig. 234 shows the force curve of the Missouri's main engines for which the data are given later on. To avoid confusion only the weights on the four cranks are used, the valves and valve gears being omitted for the present.

As in the case of a set of rotating masses, the resultant of a system of accelerating forces will in general be one force and a couple tending to turn the crank shaft at right angles to its

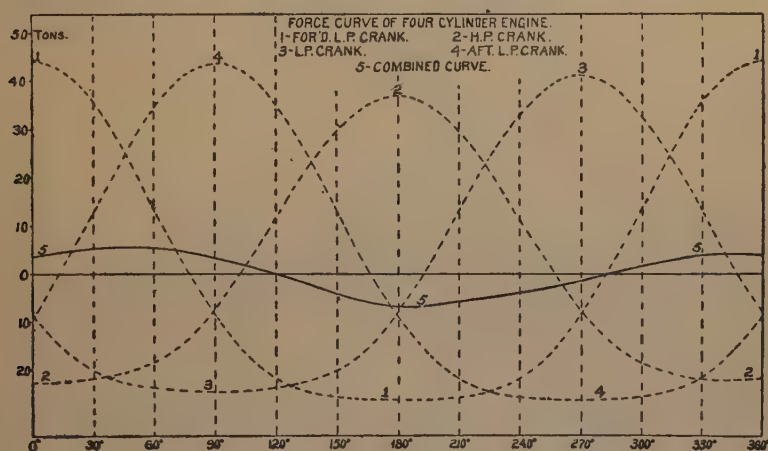


FIG. 234.

length. The couple can be found by taking moments about a radial axis in the reference plane. Thus, for each crank the moment is $Ga(\cos \theta + \frac{r}{l} \cos 2\theta)$ where a is the distance of the crank pin center from the reference plane. If the curves of force moments are laid out in proper phase for all cranks, the combined moment curve will be obtained exactly as for the force curve. From the two curves, the couple and the resultant force will then be known for every crank angle. The combined moment curve for the Missouri's engines is given in Fig. 235. Examining the two curves we have for instance at the angle 90° an unbalanced force of 7,000 pounds

and an unbalanced moment of 1,467,400 foot-pounds. Both the force and the moment are seen to vary periodically during a revolution of the shaft. They can be counterbalanced only by means of additional reciprocating weights driven off the shaft by eccentrics or cranks.

The curves shown are derived solely from the reciprocating masses, but the total vertical disturbing effects must spring also from the vertical components of the rotating forces. The simplest method of finding the combined curve for all vertical forces will be

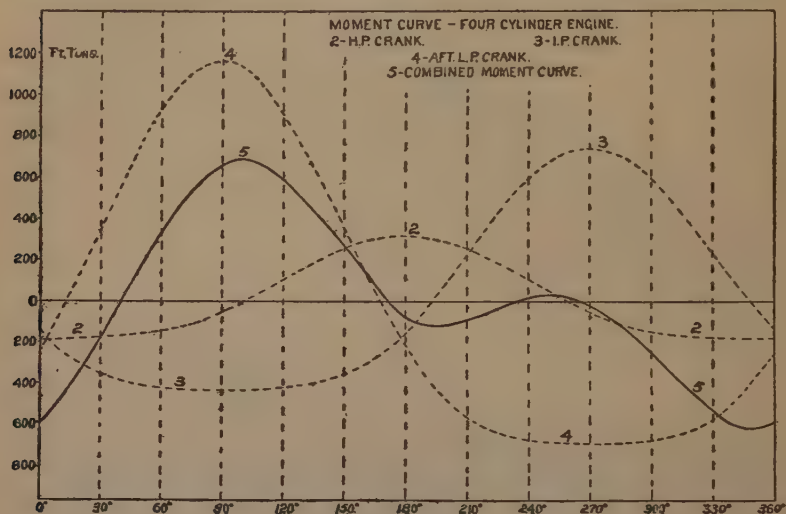


FIG. 235.

to add the equivalent weight of rotating parts to their respective reciprocating weights; then to draw the primary curves for the total weights; and to correct them for the secondary curves of the reciprocating weights. Inasmuch as all the curves are similar, it is a mere matter of scales after the curves for one unit are drawn. The lateral components of forces are neglected.

Balancing Reciprocating Parts.—The reciprocating weights are usually balanced as nearly as possible by a suitable arrangement of cranks, although additional weights, called “Bob weights,” have been used at the two ends of the crank shaft for torpedo boats. Additional balancing weights should always be reciprocating if pos-

sible, but the difficulties in the nature of mechanical construction and operation have led engineers in some cases to substitute rotating weights which neutralize half the unbalanced force and couple. The most violent forces are thereby softened down, and the new lateral forces introduced are not great enough to do much harm.

The best arrangement of cranks for balance may be studied from eight fundamental equations similar to those for rotating masses. Let W_1, W_2, W_3 , etc., Fig. 236, be the reciprocating weight reduced to the crank pin, a_1, a_2, a_3 , etc., the angles between an axial reference plane and the respective cranks, a_1, a_2, a_3 , etc., their respective dis-

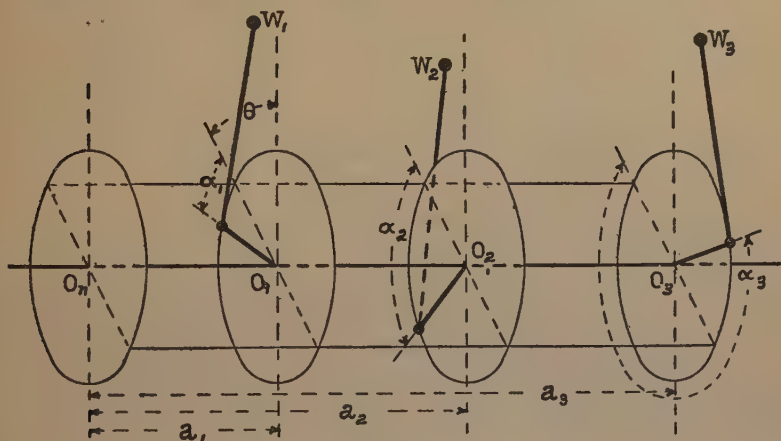


FIG. 236.

tances from a radial reference plane, and θ the angular distance moved by the axial reference plane from the plane through the center line of the cylinders. Remembering that for a single unit,

$$F = \frac{W\pi^2 N^2}{900 g} r \cos \theta + \frac{W\pi^2 N^2}{900 g} \frac{r^2}{l} \cos 2\theta$$

in which θ is the crank angle represented by $\theta + \alpha_1, \theta + \alpha_2, \theta + \alpha_3$, etc., in the above, the resultant force and moment are given in the two following equations:

$$F = \frac{\pi^2 N^2 r}{900 g} \left\{ \begin{aligned} &W_1 \cos (\theta + \alpha_1) + W_2 \cos (\theta + \alpha_2) + W_3 \cos (\theta + \alpha_3) + \text{etc.} \\ &+ \frac{W_1 r}{l} \cos 2 (\theta + \alpha_1) + \frac{W_2 r}{l} \cos 2 (\theta + \alpha_2) + \frac{W_3 r}{l} \\ &\quad \times \cos 2 (\theta + \alpha_3) + \text{etc.} \end{aligned} \right\}$$

$$M = \frac{\pi^2 N^2 r}{900 g} \left\{ \begin{aligned} &W_1 a_1 \cos (\theta + a_1) + W_2 a_2 \cos (\theta + a_2) + W_3 a_3 \cos (\theta + a_3) + \text{etc.} \\ &+ \frac{W_1 r}{l} a_1 \cos 2 (\theta + a_1) + \frac{W_2 r}{l} a_2 \cos 2 (\theta + a_2) + \frac{W_3 r}{l} a_3 \cos 2 (\theta + a_3) + \text{etc.} \end{aligned} \right\}$$

If the angles be expanded by writing $\cos (\theta + a) = \cos \theta \cos a - \sin \theta \sin a$, these equations become:

$$F = \frac{\pi^2 N^2 r}{900 g} \left\{ \begin{aligned} &(W_1 \cos a_1 + W_2 \cos a_2 + W_3 \cos a_3 + \text{etc.}) \cos \theta \\ &- (W_1 \sin a_1 + W_2 \sin a_2 + W_3 \sin a_3 + \text{etc.}) \sin \theta \\ &+ \frac{r}{l} (W_1 \cos 2a_1 + W_2 \cos 2a_2 + W_3 \cos 2a_3 + \text{etc.}) \times \cos 2\theta \\ &- \frac{r}{l} (W_1 \sin 2a_1 + W_2 \sin 2a_2 + W_3 \sin 2a_3 + \text{etc.}) \times \sin 2\theta \end{aligned} \right\}$$

$$M = \frac{\pi^2 N^2 r}{900 g} \left\{ \begin{aligned} &(W_1 a_1 \cos a_1 + W_2 a_2 \cos a_2 + W_3 a_3 \cos a_3 + \text{etc.}) \cos \theta \\ &- (W_1 a_1 \sin a_1 + W_2 a_2 \sin a_2 + W_3 a_3 \sin a_3 + \text{etc.}) \sin \theta \\ &+ \frac{r}{l} (W_1 a_1 \cos 2a_1 + W_2 a_2 \cos 2a_2 + W_3 a_3 \cos 2a_3 + \text{etc.}) \cos 2\theta \\ &- \frac{r}{l} (W_1 a_1 \sin 2a_1 + W_2 a_2 \sin 2a_2 + W_3 a_3 \sin 2a_3 + \text{etc.}) \sin 2\theta \end{aligned} \right\}$$

In case the system is balanced, both F and M are equal to zero, and all the coefficients of the θ functions must be separately equal to zero since θ may have any value. There are thus the following eight fundamental equations, simplified by selecting the reference planes in one of the crank units, so that $a_1 = 0$, $a_1 = 0$.

$$\begin{aligned} W_1 + W_2 \cos a_2 + W_3 \cos a_3 + \text{etc.} &= 0. \\ W_2 \sin a_2 + W_3 \sin a_3 + \text{etc.} &= 0. \\ W_2 a_2 \cos a_2 + W_3 a_3 \cos a_3 + \text{etc.} &= 0. \\ W_2 a_2 \sin a_2 + W_3 a_3 \sin a_3 + \text{etc.} &= 0. \\ W_1 + W_2 \cos 2a_2 + W_3 \cos 2a_3 + \text{etc.} &= 0. \\ W_2 \sin 2a_2 + W_3 \sin 2a_3 + \text{etc.} &= 0. \\ W_2 a_2 \cos 2a_2 + W_3 a_3 \cos 2a_3 + \text{etc.} &= 0. \\ W_2 a_2 \sin 2a_2 + W_3 a_3 \sin 2a_3 + \text{etc.} &= 0. \end{aligned}$$

The first four relating to primary forces are exactly like the fundamental equations for rotating masses, and the second four

containing only secondary forces are entirely due to the angularity of the connecting rods. It is not intended to imply here that the primary and secondary forces exist as separate independent forces, but simply that they are two elements into which F may be conveniently divided.

Of late, these equations have been much studied and discussed by engineers. They are of great value in working out the best arrangement of crank angles and spacing. When supplemented by the graphic method explained later, they are very easy of application, although alone they are somewhat tedious in some cases. They are best adapted to the comparison of engines already designed. For this purpose the unbalanced resultants may be separated into primary and secondary. Suppose for a given engine that none of the fundamental equations become zero and that we wish to find the axial planes and the values of forces and couples equal to the resultants. Let the planes of the resultant primary force and couple be at right angles to each other so that α_n and $\alpha_n + 90^\circ$ represent them. Let also W_n be the equivalent resultant weight, a_n its distance from the reference plane, and $W_m a_m$ the couple. Then since the couple has no component in the plane of W_n , the equations are:

$$W_n \cos \alpha_n = \Sigma W \cos \alpha$$

$$W_n \sin \alpha_n = \Sigma W \sin \alpha$$

$$W_n a_n \cos \alpha_n - W_m a_m \sin \alpha_n = \Sigma W a \cos \alpha$$

$$W_n a_n \sin \alpha_n - W_m a_m \cos \alpha_n = \Sigma W a \sin \alpha$$

From these the values of W_n , a_n , a_n and $W_m a_m$ can be obtained, and they can be substituted for the entire set of reciprocating weights, including valves, gears, and moving parts of attached air pumps, reduced previously to the crank pins.

The planes of the resultant force and couple for secondary forces may be found in the same way, simply keeping in mind that the planes for these forces are supposed to revolve twice as fast as those for primary forces, and that only weights connected with the main crossheads are counted. The auxiliary weights like valves have such long rods that angularity is not considered.

As an example of the use of these formulæ the unbalanced resultants for reciprocating weights are here worked out for the main engines of the Missouri. The weights are given in the three tables

TABLE B.
RECIPROCATING PARTS OF STARBOARD ENGINE.

Secondary Force and Moment.

U. S. S. Missouri.

No.	Location	W	$2a$	a	$W \cos 2a$	$W \sin 2a$	$Wa \cos 2a$	$Wa \sin 2a$
1	F. L. P..	8382	0	0	+8382	0	0	0
2	H. P....	7066	360°	8.71	+7066	0	+ 61544	0
3	I. P.....	7834	540°	17.83	-7834	0	-139680	0
4	A. L. P..	8343	180°	26.54	-8343	0	-221420	0
Algebraic Sum.....					- 729	0	-299556	0

$$\text{Resultant Force} = \frac{729 \pi^2 N^2 r^2}{900 gl} \text{ lbs.}$$

$$\text{Resultant Moment} = \frac{299556 \pi^2 N^2 r^2}{900 gl} \text{ ft.-lbs.}$$

$$\text{Angle of Force and Moment} = 2(\theta + 90^\circ)$$

 $\theta = \text{Angle of F. L. P. Crank.}$

TABLE C.
ROTATING PARTS, STARBOARD ENGINE.
U. S. S. Missouri.

No.	Location	W	α	a	$W \cos \alpha$	$W \sin \alpha$	$Wa \cos \alpha$	$Wa \sin \alpha$
Crank-pin Weights:								
1	F. L. P.....	4200	0	0	+4200	0	0	0
2	H. P.....	4222	180°	8.71	-4222	0	-36774	0
3	I. P.....	4210	270°	17.83	0	-4210	0	- 75063
4	A. L. P.....	4202	90°	26.54	0	+4202	0	+111520
Eccentric Weights:								
5	F. A. E.....	259	129° 30'	4.38	- 165	+ 200	- 722	+ 875
6	F. B. E.....	259	230° 30'	3.94	- 165	- 200	- 649	- 787
7	H. P. A. E..	252	303° 45'	11.98	+ 140	- 210	+ 1677	- 2510
8	H. P. B. E..	252	56° 15'	12.42	+ 140	+ 210	+ 1739	+ 2602
9	I. P. A. E..	260	323°	14.08	+ 207	- 156	+ 2924	- 2203
10	I. P. B. E...	260	217°	14.52	- 207	- 156	- 3015	- 2272
11	A. A. E.....	257	219° 30'	22.16	- 200	- 163	- 4426	- 3584
12	A. B. E.....	257	320° 30'	22.60	+ 200	- 163	+ 4514	- 3655
Algebraic Sum.....					- 72	- 646	-34732	+ 24923

$$\text{Resultant Force} = \frac{650 \pi^2 N^2 r^2}{900 g} \text{ lbs. Plane } 263^\circ\text{--}38'$$

$$\text{Resultant Couple} = \frac{37000 \pi^2 N^2 r^2}{900 g} \text{ ft.-lbs. Plane } 173^\circ\text{--}38'$$

and the sequence of cranks is shown in Fig. 237. In working out the resultants, the reference planes are chosen to pass through the axes of the forward low-pressure cylinder.

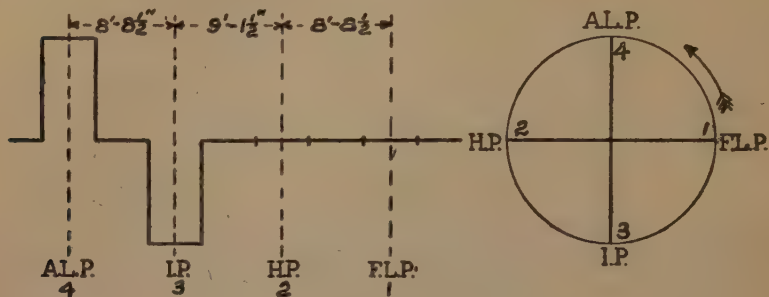


FIG. 237.

Graphic Method for Reciprocating Masses.—The reciprocating weights may be studied or balanced by the graphic method described for the rotating parts. They are treated just as if they were rotating weights placed on the crank pins or reduced to equivalent weights, as the case may be. The projections on the radial refer-

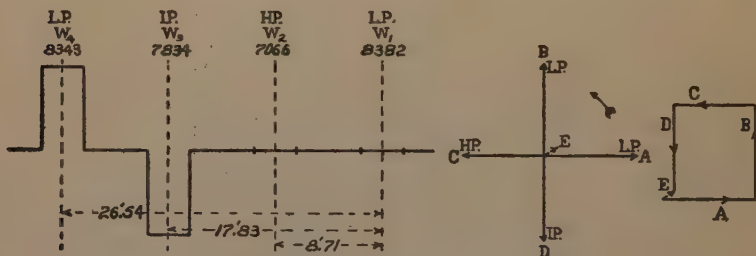


FIG. 238.

'Primary force diagram. E = Resultant force.

ence plane are made and the polygons drawn as before. There are, however, two additional polygons due to the secondary forces. One illustration, in Figs. 238, 239, 240 and 241, of the common four-crank engine with unequal reciprocating masses, is given, leaving the student to study out the process for himself. It must be

remembered that the secondary forces are laid out at twice the crank angles and that their values are the same as for the primary forces multiplied by the factor $\frac{r}{l}$. The eccentric weights are omitted for the sake of clearness.

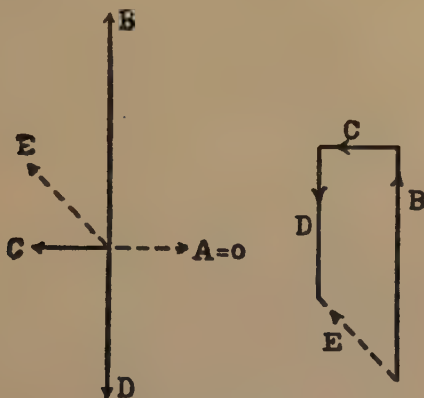


FIG. 239.

Primary moment diagram. E = Resultant moment.

It may be shown by the foregoing equations, or graphic process, that five or more cranks are necessary to procure the complete balance of an engine against all forces and couples, unless additional weights are put in. Hence, the engines ordinarily found on board

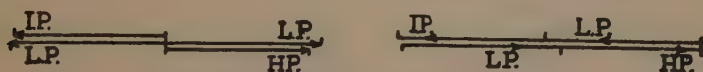


FIG. 240.

Secondary force diagram. Crank angle doubled. Resultant small.

ship are deficient in that respect. A brief discussion of the various types will be found useful.

One-Cylinder Engine.—An engine of this type is of comparatively little importance, except for auxiliary purposes, yet it can make a very disagreeable jar on board ship. It can be counterbalanced in part by weights placed at each end of the crank shaft, but the usual practice is confined to a counterbalance directly opposite each crank

as shown in Fig. 223. Where it is desirable to soften down the impulses of the unbalanced forces as much as possible, the counterbalances are made sufficiently heavy to neutralize the rotating weights and one-half the reciprocating weights. Thus, let W_1 and W_2 be the respective weights of the rotating and reciprocating parts, reduced to the crank pins; r_1 the length of the crank; W the weight of the counterbalance, and r the distance of its center of gravity

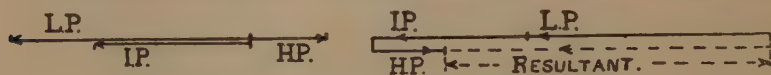


FIG. 241.

Secondary moment diagram.

from the shaft axis. Then make $Wr = \left(W_1 + \frac{W_2}{2}\right)r_1$, from which either W or r can be found. The unbalanced force when the piston is on the outboard center will then be $W_2 r_1 \left(\frac{1}{2} + \frac{r_1}{l}\right) \frac{\pi^2 N^2}{900g}$ on the inboard center $W_2 r_1 \left(\frac{1}{2} - \frac{r_1}{l}\right) \frac{\pi^2 N^2}{900g}$; and the lateral unbalanced force when the crank angle is 90° or 270° will be $(Wr - W_1 r_1) \frac{\pi^2 N^2}{900g}$. This method of balancing is thus, in effect, a reduction of the accelerating force by the introduction of a lateral force.

Two-Cylinder Engine.—This engine usually has the cranks placed 90° apart and little attention is paid to the reciprocating parts, although in some cases the cranks are placed opposite each other in different radial planes. The latter arrangement enables the engineer to balance the primary forces and the centrifugal forces by making the weights of all moving parts alike, but there still remains the secondary forces and the couples tending to turn the shaft end for end. This type of engine is not used for propelling purposes and but little for auxiliary purposes. Its turning moment is no better than that of a tandem, or single-cylinder, engine.

If the cranks are 90° apart, the primary, secondary and centrifugal forces can be counterbalanced by weights at the two ends of the crank shaft, but the upending couples still remain. It is, on the whole, better to counterbalance each unit separately, exactly as for the single cylinder. The most that is commonly done is to balance

the rotating weights and leave the reciprocating parts to take care of themselves.

A new arrangement of two cylinders has been suggested by Mr. McAlpine for the purpose of balancing all forces and upending couples. The two cylinders are placed in the same athwartship plane and connected by means of an oscillating lever to the same crank pin, as shown in Fig. 242. This arrangement is a very good solution of the problem and the lateral couple which it introduces is

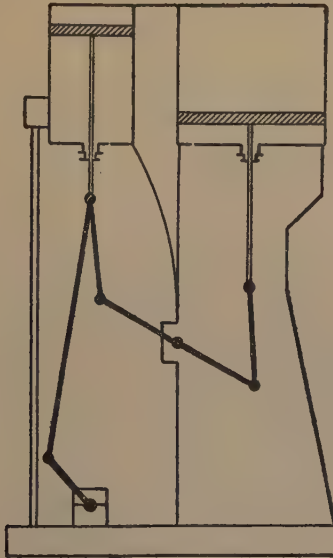


FIG. 242.

not seriously objectionable. Its turning moment is, however, no better than that of a single-cylinder engine. This objection can be removed by two units, thus having four cylinders on two cranks set 90° apart. The piston of the outboard cylinder and the connections to the cylinder directly over the crank shaft correspond to the attached air pump run from the crosshead. The latter is generally too small to have much effect on the balance, although it suggests a method of adding weights for balancing one or more cylinders.

Three-Cylinder Engine.—The usual type of three-cylinder engine has three cranks set 120° apart. As already shown for rotating weights, a complete balance of simple forces can be accomplished. The same thing is true for the reciprocating parts if they all weigh the same amount, but couples will still exist. This can be shown by substituting in the fundamental equations, as before. While much can be done for this type of engine by the addition of rotating, or bob weights, little attention is usually paid to balancing the couples.

Four-Cylinder Engine.—This engine has developed into the normal type for propulsion in the navy. Most often, it is in effect a triple-expansion engine with two low-pressure cylinders, each cylinder having its own crank. As usually designed with cranks at 90° and reciprocating weights uniform, there is a complete balance of primary, secondary and centrifugal forces, but considerable couples remain.

The Bureau of Steam Engineering has for many years adopted this system of balancing, generally making the weights on opposite cranks the same. To this end the larger piston is designed first, and then the smaller piston is dimensioned to give it the same weight. The data for the engines of the Missouri show, however, that other considerations seemed of more importance than accurate balance, for the reciprocating weights vary from 7066 pounds in the H. P. cylinder to 8382 pounds in the L. P. cylinder. The resultant forces are not great enough to affect the hull of the ship much, as she ran with little vibration on her trial trip.

One case of the four-cylinder engine is worked out here for the purpose of illustrating the use of the fundamental formulæ. Let $W_1 = W_2 = W_3 = W_4 = W$.

$$\alpha_1 = 0, \alpha_2 = 180^\circ, \alpha_3 = 90^\circ, \alpha_4 = 270^\circ.$$

The equations for primary and secondary forces then becomes

$$W + W \cos 180^\circ + W \cos 90^\circ + W \cos 270^\circ = 0.$$

$$W + W \cos 360^\circ + W \cos 180^\circ + W \cos 540^\circ = 0.$$

$$W \sin 180^\circ + W \sin 90^\circ + W \sin 270^\circ = 0.$$

$$W \sin 360^\circ + W \sin 180^\circ + W \sin 540^\circ = 0.$$

Hence all simple forces are balanced.

The equations for couples are:

$$Wa_2 \cos 180^\circ + Wa_3 \cos 90^\circ + Wa_4 \cos 270^\circ = -Wa_2.$$

$$Wa_2 \sin 180^\circ + Wa_3 \sin 90^\circ + Wa_4 \sin 270^\circ = W(a_3 - a_4).$$

$$Wa_2 \cos 360^\circ + Wa_3 \cos 180^\circ + Wa_4 \cos 540^\circ = W(a_2 - a_3 - a_4).$$

$$Wa_2 \sin 360^\circ + Wa_3 \sin 180^\circ + Wa_4 \sin 540^\circ = 0.$$

Combining the first two of these, and putting in the constant factor $\frac{\pi^2 N^2 r}{900 g}$ and we get for the greatest primary force couple

$$M = \frac{\pi^2 N^2 r}{900 g} W (a_2^2 + (a_3 - a_4)^2)^{\frac{1}{2}}.$$

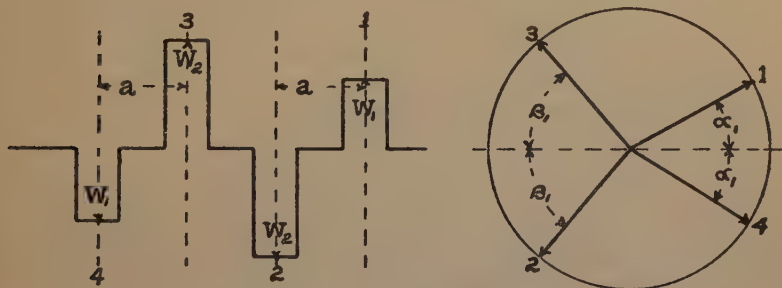


FIG. 243.

From the third equation the greatest secondary force couple is found to be

$$M = \frac{\pi^2 N^2 r^2}{900 g l} W (a_2 - a_3 - a_4).$$

The planes of these couples can be found most readily by the graphic method.

By a study of the formula, other arrangements of the cranks have been found to get rid of the primary couple as well as the simple forces. One of these is the Yarrow-Schlick-Tweedy system of balancing. This is essentially a method of solving the fundamental equations by throwing out the two containing secondary force moments. There remain, then, six equations which can be simplified by imposing certain conditions, as, for instance, that the spacing of the crank shall be symmetrical about a radial plane midway between the two inside cranks, and that the reciprocating parts of the two outside cylinders shall weigh the same.

The case is best illustrated by the solution given in Fig. 243. The cranks are arranged in pairs, each pair having reciprocating parts of the same weight. Numbering the cylinders from forward, Nos. 1 and 4 form one pair, and Nos. 2 and 3 the other. An axial plane which bisects the angle between the cranks 1 and 4 also bisects the angle between 2 and 3. The axial spacing of 1 and 2 is the same as that of 3 and 4. The actual value of W_1 and W_4 is determined by the design; then the values of W_2 , W_3 are derived from the formulæ. The space a_2 is also given in the design and a_3 , a_4 are found.

The full analysis of this system of balancing is beyond the scope of this chapter. It will be found in the transactions of the British Naval Architects for 1900. The turning moment on the shaft does not appear to suffer by the variation of the crank angles as above, if care is exercised in the distribution of pressures in the cylinder. It may even be improved; yet engineers are not agreed upon its superiority over the ordinary four-cylinder engine with cranks at 90° . The secondary force couple may be large and if the engine is not well placed in the ship the vibrations may be serious.

Five- and Six-Crank Engine.—Both of these types have been proposed as a means of balancing the marine engine in practically all respects. They are not yet in common use and the complication of so many parts seems likely to retard their development except for very high rotative speed. The best arrangement for a six-crank engine is to regard it as composed of two similar three-crank engines placed end to end with crank sequences reversed. If both sets of cranks are balanced independently for forces, the couples being the reverse of each other, will form a complete balance. If the cranks be numbered from forward, then 1 and 6, 2 and 5, 3 and 4, are respectively in the same axial plane and point in the same direction, as shown in Fig 244. It is not necessary that the parts should weigh the same, as they can easily be adjusted to balance one another by a study of the fundamental equations.

The five-crank engine is the same as the six with the two middle cranks squeezed into one, thus making the parts of the middle cylinder considerably heavier than those for the other units. The reciprocating parts of this engine can be perfectly balanced by a careful

spacing of the cranks, but the rotating parts cannot be balanced against one another unless they vary in weight. Inasmuch as for obvious reasons they are made the same, this type is not as perfect as the six-crank engine.

Steam Turbine Balance.—The steam turbine which is coming into use for marine purposes requires very careful balancing. All the parts are rotative, but each part should be separately balanced. The speed is so high that a very small excess of weight on one side of the shaft may develop great pressures on the bearings or serious strain on the shaft. A weight of one ounce placed on the rim of a turbine disc four feet in diameter making 1200 revolutions per minute causes a lateral pressure of 62 pounds; and the same disc if $\frac{1}{100}$ inch out of true would generate a lateral pressure of 400 pounds. If a

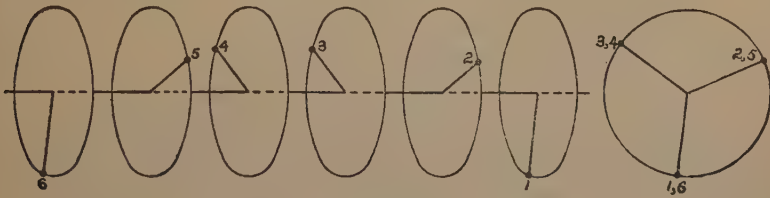


FIG. 244.

Sequence of cranks for six-crank engine.

1-pound bolt be put near the rim on either flat surface there is a centrifugal moment of 750 pounds $\times$ the half thickness of the disc. It is so necessary that all centrifugal forces and couples should be eliminated, that the ordinary statical balance is not always sufficient. The turbine is sometimes run under service conditions and the deflection of the shaft is carefully examined. Small weights are then added or taken off where necessary.

Location of Engines to Reduce Vibration.—It is often possible to neutralize the vibratory effect of unbalanced forces and couples by care in locating the engine in the ship. The vibration of the hull takes place about two nodes. The after node is not far from one-fourth the length of the ship from the stern. Unbalanced free forces would not be felt if the engine were located in the node, but a force couple would be serious in its effect if its period corre-

sponded to that of the ship. On the other hand, if the engine is placed some distance from the node the free forces become effective in setting up vibration and the couples ineffective. An adequate study of this subject would enable an engineer to place an unbalanced engine in a ship without serious results. The chief difficulty lies, however, in locating the nodal points before the vessel is built and loaded. Instruments have been devised to measure the amplitude and number of vibrations on board ship, and numerous experiments have been made to obtain data for calculating them, but the subject still remains obscure. The most satisfactory method of dealing with the problem of vibration is to balance all machines as completely as the demands of good practical working will permit.

CHAPTER XVI.

OPERATION AND MANAGEMENT.

Fire Room Management.—The following is a brief outline regarding the routine of operation and management in the fire room, in raising steam in the boilers and getting underway. It is assumed that the boilers are of the water-tube type, and that they are empty.

Raising Steam.—Previous to the time announced for starting fires a general examination of the fire rooms and boilers should be made to be sure that everything is in readiness for getting up steam. The following are among the more important points to be looked after:

Examine grate and bearer bars, brick work in the boilers, the furnace and cleaning doors, repairing or renewing where necessary. See that fire tools, ash and coal handling gear are in good condition; that the bilges are free and clear of obstructions to flow of water to the bilge wells; see strainers clear and that no inflammable material is in close proximity to boilers. If oil is carried as fuel see that there are no leaks about the piping or tanks, and that the burners are in good condition.

Before starting fires see that there is a sufficient supply of feed water available, and that all instructions contained in the following Naval Instructions are carried out:

“Before starting fires in any boiler, all drain cocks, bottom and surface blow valves, handhole and manhole plates, and other valves and fittings not intended to remain open, shall be examined to see that they are tightly closed. The safety valves, boiler stop valves, feed check and stop valves, and water column valves shall be examined, and the water gauge and test cocks shall be tested, to see that they are all in proper working order. It shall be definitely ascertained that the valves and pipes leading to the pressure gauges are wide open. The water shall be brought to a height that is

slightly below the normal steaming level. The air cock shall remain open while the water is being run down and while steam is being raised, and shall be closed after steam has formed.

“While steam is being raised in a boiler, close attention shall be given to all the boiler fittings and feed arrangements to insure that they are in all respects in proper working order. Special care shall be taken, in setting up the nuts of handhole and manhole fittings, that no greater leverage is applied than that afforded by the proper spanner provided for that purpose. Ash pit doors of automatic or balanced type shall always be left mounted while boilers are under steam. The boilers shall be connected to the steam line only when there is a difference of pressure not exceeding ten pounds, and the boiler stop valve shall at first be only slightly opened, to allow the pressures in the steam line and in the boiler to equalize gradually. After the pressures have equalized, the stop valve shall be further opened gradually to such extent as required.

“Whenever steam is raised in a boiler, in order to insure that the safety valves are in good working order and to ascertain the exact pressure at which they will lift, the steam pressure shall be allowed to rise until these valves should lift if properly adjusted. This may be done after the boiler has been connected, if more convenient, but the boiler shall not be continued in use unless the safety valves have been correctly adjusted.”

Slack smoke pipe guys, see that gage lights are in order, close furnace and ash pit doors of boilers not in use, and see that their main feed and check valves are closed. During this preparation the coal passers have placed several buckets of lumpy coal in front of the furnaces. The grate is now covered with a layer of coal from three to six inches thick, extending to within a foot of the door. In the furnace mouth, coal and broken wood are piled and lighted off with a shovelful of live coals from an auxiliary boiler, if one is in use; if not, it is touched off by lighting old oily waste placed just within the furnace.

See that the ash hose is rigged in place, that all loose gear that might be carried away in a sea-way is properly stowed and secured, and that the cocks and valves about the boilers have been moved and are in working condition.

While the fires are burning up the furnace doors are left slightly ajar and the ash pit doors partly closed. As the coal becomes ignited, it is worked back from time to time with the hoe, mixed with the coal at the back, and fresh fuel is put on in front. Regular rounds of coal are brought out by the coal passers at regular intervals until the end of the run.

As soon as all the coal on the grate is ignited, the furnace doors are closed and the ash pit doors opened wide. Keep the fires leveled off, sprinkle with fresh coal from time to time, and shove the fires back from the furnace doors, to prevent burning the linings.

Since steam may form at widely different times in different boilers, owing to the difference in draught, firing, etc., the temperature of the water in the boiler should be taken at intervals and the fires checked or urged, so that the rise in temperature of the different boilers may be as uniform as possible.

The formation of steam will be shown by the escape of vapor from the test cocks, which are then to be shut off. In case no auxiliary steam pipe is fitted, the safety valves are closed, as soon as all air is judged to have escaped, and the steam is then allowed to rise to its maximum and each boiler in turn is connected to the main steam pipe as its pressure rises to that in the pipe. As the pressure rises a few pounds above the atmospheric pressure, close all drains which are blowing steam, drain steam drums, separators and main steam pipes, close their valves, and open wide the boiler stop valves. The smoke pipe guys previously loosened may now be adjusted to the increased temperature of the smoke pipe, to properly support it in a sea-way.

While steam is rising in pressure a careful examination should be made for leaks, especially around joints, man and handhole plates, etc. If anything serious is discovered, such as would soon place the boiler out of commission, start fires in another boiler at once.

The auxiliary machinery in the fire room should be examined while raising steam. If there is steam on an auxiliary boiler, try the feed pumps and ash hoists and see that bilge strainers are clear. If there is no auxiliary steam, try the fire room auxiliaries as soon as possible after steam is obtained on the main boilers.

For water-tube boilers the average time required for steam is about two hours, but in an emergency the necessary time would be that required for warming the engines; steam can be raised in some of these boilers within thirty minutes.

All machinery and piping must be gradually heated and care taken to prevent water rams or unequal expansion. The following precautions required by Naval Instructions must be observed:

“The cylinders, receivers, and steam jackets of all engines shall be gradually and thoroughly heated before steam of full pressure is admitted to them. Whenever practicable the connections between the boilers and main engines shall be opened as soon as fires are lighted to allow the hot air and steam as it rises to circulate through the engines and warm them gradually.

“Water must not be allowed to accumulate in the receivers or jackets of engines, but care shall be taken that steam is not blown through the traps and wasted while the engines are working.

“The greatest care shall be exercised to guard against water rams by carefully draining all pipes or other spaces where any water might accumulate before steam is admitted to them. In opening the valves connecting the boilers to the steam pipes or in connecting different sections of steam piping, the by-pass valves, if fitted, shall be opened first, and the pipes warmed and the pressure equalized before the large valves are opened. The drains shall be kept open until it is certain that the pipes are entirely free of water.”

Control of Steam.—The management of the steam as the pressure rises to its limit, or working pressure, requires great skill and care. The tubes and headers must not be unnecessarily exposed to sudden or unequal cooling by opening casing or furnace doors or by pumping cold water into boilers; the dampers may be closed and ash pit partially closed if desired to retard raising of steam; if these do not answer start all auxiliaries that may be used to advantage, *i. e.*, evaporators, bilge pumps, ice machines, etc. Bleeders may be opened, but care must be taken that they are not opened too suddenly or too wide, in order to prevent damage to the condenser tubes. As a last resort the excess pressure may be taken care of by the safety valves, but this is bad practice as it entails a loss of water, and

the noise is at times a serious hindrance to performance of duty on deck.

Standing By.—If the time of starting is postponed for two or three hours, the fires may remain spread and kept lightly coaled, with evaporators and auxiliaries running. If some such precaution is not taken, the efforts to keep the steam down will have so deadened the fires that after starting the main engine the steam will fall rapidly, and with difficulty be brought up to the normal pressure.

Banked Fires.—The following requirement of Naval Instructions should be observed:

“Banked fires shall not be kept in water-tube boilers; nor shall heavy banked fires be kept in any other type of boilers except in emergencies, and in such cases ash pit doors shall not be closed. When the main engines are not to be used for twenty-four hours or more, fires shall be allowed to die out in all water-tube boilers, except those needed for auxiliary purposes, and when the main engines are to be used again within twenty-four hours light fires shall be kept spread in such boilers.”

In the Engine Room.—While the preceding operations are under way in the fire room, the engine room force will be engaged in the preparatory work about the main engines and auxiliaries. A general examination should be made to see that all fastenings, nuts, set screws, etc., especially small bolts about the working parts of the main engines, air and circulating pumps, are tightly secured. The work of getting ready is divided among machinist's mates, oilers and engine room helpers. See that the lubricating system is in order, cups filled, wicks in condition and oil pipes clear. Ease off bulkhead and communication and regulating stop valves to engine. Proceed to warm up engines as steam forms, by means of the steam jacket, admission and drain valves. Open up all drains on the main engines. Open outboard discharge from the condenser and the main injection and start the main circulating and air pumps as soon as steam is available. Try water service and see that pipes are clear.

If forced lubrication is used, see that the oil pumps are tested, that oil has free discharge to and from bearings and that the joints in the casing around the engines are properly secured and tight to prevent waste of oil.

In starting any auxiliary machinery, proper care must be taken to see that the steam cylinders are freed from water with the aid of the cylinder relief valves, or drains.

Examine crank pits and clear away all obstructions, remove all gaskets placed to keep dirt out of the bearings and any other temporary coverings. See that the jacking engine is disconnected, and that all gear about the engine rooms and shaft alleys is secured for sea, and that there is nothing loose likely to get into contact with the working parts of the machinery when the ship rolls. Examine bilge pumps and bilge injection strainers to see that they are clear; warm up and try the reversing engine and run the links back and forth. Turn steam on the capstan and steering engines and try them; also try the siren, steam whistle, engine telegraphs, voice pipes, and gongs. Slack back the stern glands so that water will leak through slightly when underway. Open boiler stop, bulkhead and cross communication valves and main stop valves the required amount for running. When the steam pressure is sufficiently high, make a final inspection to see that the men are at their stations. If there have been any adjustments or overhauling on the main engines since last in operation, they should be jacked over one or two turns to see that everything is clear and in working order.

The above procedure having been carried out properly, steam raised to the required pressure, the proper water level in the boilers, the feed pumps and feed system satisfactory, the cylinders warmed and drained and wicks in, obtain permission from the officer of the deck to turn the engines, in order to see that they are in proper condition for working in ahead and astern directions.

On obtaining permission, turn the engines slowly two or three turns in each direction alternately, and thus avoid putting any way on the ship; repeat this several times until the engine starts in either direction promptly and the cylinders are free of water.

The main stop valve being opened, the throttle is opened slightly and the links thrown over; if the engine does not start off, work

the links back and forth, and, working the throttle gradually, the engine will start in one direction or the other. Owing to the water in the cylinders the engine may not pass the centers. In this case the water is gradually worked out through the cylinder relief and drain valves, the links being moved from one gear to the other, and the engine "rocked"; the pistons will gradually get nearer the ends of the stroke, and finally, when the water is cleared, the cranks will pass over the centers and continue the revolution. If the high-pressure engine is on a center the engines can be started with the aid of the *by-pass valves* to the receivers. There should never be any difficulty in starting, however, even with the high pressure on a center, unless in an emergency, when there is insufficient time to warm the engines; the by-pass valves should never be used unless absolutely necessary. The engines can be thoroughly tried without putting any appreciable strain on the moorings by thus alternately turning them.

Note to what degree the main condenser retains its vacuum after stopping, which will give an indication as to its condition regarding air leaks. Shut the main engine throttle valve and put the links in mid-position. The engines should now be reported ready. Inspect for leaks about the engines and fire rooms and see that the auxiliaries are working properly; get a pump ready to start on the fire main promptly for washing down the anchor and chain.

The time when the ship is to get underway naturally is known by the senior engineer officer and the above preparations are started in ample time, so that soon after the engines are tried and reported ready, the signal for getting underway may be expected. After all is ready, should the time for starting be postponed for several hours, take out the oil wicks, shut steam off the capstan and steering engines and slow down the air and circulating pumps; if the fires are under complete control, and the bleeders are not likely to be required, these pumps may be stopped and the outboard valves closed.

DUTIES OF A STEAMING WATCH—FIRE ROOMS.

Management of Fires Underway.—Keep a uniform pressure of steam regardless of the varying speed of the engine, and at the same time economize coal. Fire regularly and evenly. As the

boilers are in separate compartments, and often the main engines can neither be seen nor heard, it is advisable to provide some speed indicator or simple apparatus by means of which the water-tender will be kept informed of every change, so that he may know at once if fluctuations of steam are due to variations of speed. He should also be informed in advance, when possible, of proposed changes.

Firing a Furnace.—This subject is dealt with fully in the text-book on Boilers.

Cleaning Fires.—If there is much slag or clinker on the bars, it will be necessary to give the fires a thorough cleaning. As a rule fires will do without cleaning for about twenty-four hours, so that on runs not exceeding a day no thorough cleaning will be required. If there is any chance of the run being longer, the cleaning should begin about twelve hours after the fires are started, one-third of the fires being taken each watch.

The fronts are marked 1, 2 and 3 to show which furnaces are to be cleaned by the respective watches. The fires to be cleaned by any one watch should be selected as far apart as possible. Before beginning, report the necessity to the engine room. Fires should not be heavily coaled just before cleaning, nor allowed to burn too low; the most capable firemen will clean their fires when but little burned down, having a fairly heavy layer of good white fire to commence with. In this way formation of steam is but little checked. A less skillful fireman may find it necessary to allow fires to burn lower, and steam pressure is liable to suffer during his cleaning.

In the meantime, get the necessary tools ready, the ash hose led out, and remove all coal from before the furnaces. Begin by closing the damper, then with the hoe shove to one side all the good fire, leaving the slag and clinker exposed; dislodge the latter with slice bar, and remove it with the hoe, thereupon it is immediately wet down by a coal passer. Now haul the good fire to the clean side, leaving the slag and clinker at the other side bare. Remove the clinker as before. Finally spread the good fire evenly over the whole surface of the grate. Then sprinkle the whole with fresh coal, open the damper, and haul ash pans. This subject is treated at length in the text-book on Boilers.

Sweeping Tubes.—After steaming for about twenty-four hours soot will begin to accumulate on the fire side of the tubes. This reduces the economy of the boiler, as soot is a poor conductor of heat, and it interferes, too, with the draft, and causes considerable difficulty to be experienced in keeping steam up to its normal working pressure, especially when the maximum speed for the number of boilers in use is required. Consequently the necessity for regularly and systematically cleaning tubes is apparent. Cleaning and dusting doors are fitted in the boiler casing through which a pipe (called a lance) carrying steam or air under pressure is introduced, which dislodges the soot, and, aided by the draft, carries it up and out of the smoke stack. Sometimes the blowers are run slowly to assist the draft during this operation. Steam should never be used for sweeping tubes when a boiler is cold, as condensation causes moisture to be deposited on the tubes, causing rust to form and the caking of what soot is left, or which may eventually be deposited. As sweeping or blowing tubes has a tendency to cool the tubes and thereby check the formation of steam, but one side of a boiler should be cleaned at a time. In some recent boilers, perforated pipes are permanently led to those localities where the soot is most likely to collect, through which steam or compressed air or both can be turned on at will by means of valves fitted for that purpose.

Routine of the Watch at Sea.—The routine is about as follows: Begin cleaning fires about fifteen or twenty minutes after the watch comes on and continue at intervals until all the fires assigned to the watch are cleaned, which will be about six bells. After each fire is cleaned it must be allowed to burn up and get into condition for making steam. After this it is useless to wait any longer before beginning to clean the next fire. Naturally, the intervals of time between the cleaning of different fires will vary according to the amount of fresh coal required to get the fire up to its original thickness. As soon as the fires are all cleaned, the ashes wet down and heaped near the ash hoist, report to the officer of the deck through the engine room that ashes are ready for hoisting. Several deck hands will then be detailed to carry the buckets from the ash hoist on deck to the ship's side and dump them. A fireman is, in the meantime, detailed to get the ash engine ready, and the coal passers stand

by to fill ash buckets. If ash ejectors or expellers are used the procedure for hoisting ashes is eliminated, as the ashes can be discharged overboard when convenient for the fire room force without assistance from the deck divisions. During the whole watch a steady feed, both with regard to quantity and temperature, is to be kept on all boilers in use. The fact of the check valves working is ascertained by putting the ear to the valve chamber. That portion of the feed pipe between the check valve and the boiler should not be hotter than the feed water. If the amount of water in the gage is greater than it should be, judging from the normal feed and known opening of the check, the cause should be inquired into at once. When the boilers are placed in an athwartship direction and the ship has a list to port, the water must be carried low in the front gages of port boilers and high in front gages of starboard boilers, and vice versa. If oil appears in the gages, it is an indication that the surface blow must be used. Every effort must be used to keep oil from getting into the boilers. If test shows water in any or all the boilers to contain salt, immediate steps are taken to ascertain the cause, as it is of the utmost importance that water-tube boilers be kept free of salt. Close observation of the boilers is necessary when salt is present to prevent priming. If the cause of salt being present is not of a nature prohibiting immediate remedy it should be rectified at once, and then the bottom blow may be used by opening wide for a few seconds only, and at intervals of not less than four hours. Naval Instructions concerning this subject read as follows:

“Any indications of the presence of sea water in the feed tanks, or of considerable increase in salinity of the water in the boilers, shall be investigated promptly and the defect corrected with the least possible delay. As a rule, when sea water becomes unavoidably admitted to the boilers, blowing down shall not be resorted to until the salinity exceeds six hundred grains of chlorine per gallon in Babcock and Wilcox or fire-tube boilers, three hundred grains in Niclausse boilers, and one hundred and fifty grains in small bent-tube or torpedo-boat boilers. When bottom blow valves are used to reduce the salinity of boilers under steam, they shall be quickly opened wide for a few seconds only, and at intervals of not less than

four hours. Too frequent use of these valves shall be avoided, as it causes them to leak, indirectly wastes fuel, and promotes corrosion.

“When it is required to empty the boilers, they shall not be blown down, as this practice is liable to cause leaky tubes and joints, but the water shall be allowed to remain until it becomes cool, and shall then be either pumped or drained out.”

Where there may be a great deal of dirt in the water it will eventually collect in the mud drums or boxes, and this is also removed by the bottom blow. Intelligent use of the bottom blow is essential to the safety of the boiler, experience alone teaching when and how much it should be used. Before using the bottom blow the water level should be raised to about the third gauge cock and it should never be lowered out of sight in the gauge glass. The water gages must be blown through occasionally to clear them out, and the test cocks used frequently as a check on them. If there is much difference in the pressure indicated by steam gages of different boilers, the cause must be discovered. It sometimes happens that stop valves become partly closed by accident.

The thickness of the fire must be kept proportioned to the strength of the draught. Increase of draught has a tendency to raise the water level and a decrease to lower it; therefore, when working under strong forced draught, when it is desired to reduce speed, half an hour's notice should be given, the draught gradually reduced during that time and more water pumped in. If necessary to stop the engines suddenly, get rid of surplus steam with bleeders or safety valves, or both. On the other hand, half an hour's notice should be given before putting the blowers on full. In the meantime, fires must be built up thick and water level reduced, otherwise air will be blown through the grates and water may be lifted so high as to go over into the engine. Coal must be used equally from both sides of the vessel and taken as it comes, not picked over.

In accordance with Naval Instructions daily and weekly tests of all feed water and water in each boiler are made to ascertain the acidity, alkalinity and salinity; the method of making these tests is described in the text-book on Boilers.

The double bottoms in fire rooms must be sounded, temperature of fire room and bunkers taken every watch, the regulations with regard to the use of lamps in bunkers enforced, a close watch must be kept on water in the bilge, and the strainers must be kept clear. At sea make a practice of keeping all water-tight doors in bunkers closed, as far as practicable.

Starting an Additional Boiler.—Proceed as before described for getting up steam, except in this case the main safety valve must be raised or air cock opened to allow the escape of hot air, and closed when steam forms. The boiler stop valve cannot be eased off its seat, so if it is a globe valve it will probably stick fast when hot. When it is desired to open this valve, slack back the nut on the yoke about half a turn, open the valve a very little, then screw up the nut. Great care must be taken not to connect up this boiler until the fires have thoroughly burned up and are in good condition to make steam, and the pressure is equal to or greater than that on the other boilers. Drain the branch pipe between the boiler and stop valve, also stop valve chamber. Do not open the valve wide at first, but unseat it until steam is heard to pass through, then leave it alone for a few minutes before opening to normal width. While this is going on watch the index of the steam gage closely to observe if there are any violent fluctuations. Also, watch the water gage to see if there is any tendency to lift water. If there is, shut off the stop valve partly or wholly until these fluctuations cease.

To Disconnect a Boiler.—Allow the fires to burn down, keeping them leveled off. When it is judged that they will make no more steam, close the stop valve and dampers, partly close ash pit doors, and shut off feed. When fires have almost burned out, close ash pit doors completely. If steam should rise above the limit, the boiler stop valve may be opened for a minute or two, then closed. If there is any evidence of the presence of oil in the boiler, the surface blow should be used to get rid of it, so that when the boiler is finally emptied a layer of oil will not be deposited on the interior. When fires are out and the boilers cold the furnaces must be hauled, and may then be primed if there is any chance of the boiler being

used again soon. Before priming be sure that the grate bars are not hot enough to set the coal on fire and that there is sufficient water in the boiler. If not to be used for some time do not prime furnaces, but fill the boilers with water to the air cock, making sure that all air is expelled.

Preparations for Coming Into Port.—Upon receiving notice that the ship will come to anchor at a given time, regulate the firing so that the fires may burn down as much as possible, having due regard to the necessity of keeping steam to maneuver the ship. Make an exact estimate of the amount of coal on hand, allowing for what will be required to reach port. If there is no auxiliary boiler, select one main boiler to be used in port, choosing one that has been used least, provided it does not require repairs, and is not adjacent to one which must be entered, and provided also that steam connections are such that it may be kept connected with the auxiliary steam pipe service, while making repairs to leaky joints and sections of pipe, or overhauling boiler stop valves that may require such attention. Make an inspection to see if the list of repairs reported to be necessary is correct and complete. Look at the escape pipes to see if any of the main safety valves are leaking, also inspect steam whistle and siren. If there is more coal out than is necessary, return it to the bunker or heap it in front of the boiler which is to be used in port. Shut any additional watertight doors in bunkers that may be found unnecessarily open. Send up all ashes without regard to the hour. If there is an auxiliary boiler to be used instead of a main boiler, start fires in it in time to allow steam to form and rise slowly to the required pressure, by the time that steam on the main boilers is used up, which will be about an hour or two after arriving in port. Have coal ready to send up for use in the steam launch.

Arrival in Port.—Upon receiving word that the engines will no longer be required, the fires are allowed to die out in all except the boiler to be used for auxiliary purposes, the ash pit doors partly closed and dampers wholly shut. The time required for the boilers to cool will vary with their size and other conditions. If possible, work on the boilers should be postponed until they have cooled off.

If for any reason it becomes necessary to blow down a boiler at once, shut it off from the other boilers, raise the safety valves until the steam pressure has fallen to about 25 or 30 pounds in excess of pressure of water overboard at the level of the bottom blow. Blowing down will then cause less jar and shock to boilers and pipes.

While waiting for the boilers to cool, the fire tools may be repaired, slice bars straightened, new heads put on hoes, and shovels trimmed. Work on bunkers, etc., may be done. If there is no work to be done on a boiler which necessitates pumping it out, if the water is fresh and not greasy, it should be left in.

Getting Underway—In the Engine Room.—A machinist's mate, under supervision of the officer of the watch, should stand by the throttle and reversing gear; a reliable man at the engine room telegraphs and voice tubes, and a machinist's mate or oiler at the feed and filter tanks to keep height of water in same under control when the engines first get underway. The engines are operated in obedience to orders received by means of the engine room telegraphs, gongs or voice pipes. When underway, all parts of main and auxiliary machinery that have been overhauled or adjusted since last operation should be given special and close attention until satisfied that they are in good working condition.

After the engines are fairly underway the following adjustments should be made:

1. The cut-offs, if previously run out to facilitate handling the engines, should be adjusted for economy, smooth running and equalization of power by the independent linking-up gear.

2. Regulate the main injection for the proper quantity of circulating water.

3. Adjust steam jacket pressure and see that jacket drains are opened wide.

4. Regulate the water service.

5. Make a general inspection of the engine and fire rooms, taking care to see that bilge pumps are working properly and strainers clear; feed pumps working well; and inspect the fires for proper thickness, the glass water gages, and try the test cocks. In the engine room inspect the bearings and see that all are running cool.

6. See that all the particulars of starting, stopping and previous

preparations for getting underway are correctly entered in the steam log. Attention should be paid to the engine room gages to see that the proper pressures are maintained, and the proper vacuum in the condenser.

In the adjustment of the independent linking-up gear, experiment should be made while underway in order to determine the best distribution from the point of view of economy and smooth working.

Indicator cards should be taken to see that all is going on in the cylinders as intended in making adjustments. See that a proper amount of feed water supply is maintained by observations of the water level in the feed tank. If additional water is required above that supplied by the evaporators, it is obtained from the reserve tanks or double bottom compartments. Any overflow of the feed tanks should be promptly investigated; it may be due to failure of the feed pumps, or feed valves closed or insufficiently opened.

After the engines are started, keep them going to the end of the run. This requires a close watch to anticipate and prevent any accident or slacking back of fastenings.

While the engines are running have a careful watch kept on gages, thermometers, counters, stuffing boxes, bearings, etc., and the attendants accustomed to the normal sounds made by the various working parts. Changes in the indications of gages and variations in the normal sounds must be investigated, and the causes ascertained.

If there is any doubt about a thermometer reading, test it by another in the same place.

If there is a slight dew on the brasses, they cannot be hot.

If there is a lather formed at the ends of the journals, they are properly lubricated and cool. If oil runs out clear and unchanged in appearance, the supply is too great; this applies where forced lubrication is not used.

A great deal of information with regard to the running of an engine may be obtained by the sense of hearing. First, the sounds may be divided into two general classes, the normal and the abnormal. Among the former may be mentioned a leading thump of one of the crank pin brasses as the pin passes one dead point. It rarely happens that all the crank pin brasses are so evenly adjusted

that some journal does not make more noise than another. Then there is the slip of the link block in the link, the steam flowing into the valve chest, exhaust into the condenser, the valves of the different pumps, the periodic blowing of the trap, etc. The abnormal sounds will be treated of later.

The two foregoing general classes may be divided into sounds which are coincident with the motion of the main engines and which, therefore, must come from some part connected with them, and, on the other hand, sounds not coincident. These two latter classes may be further divided into the light, tinkling sounds which must obviously be made by looseness of some light piece, and the dull, heavy thuds which come from heavier parts. A good ear can also distinguish between a clear, bell-like sound given out by solid brass and the less musical note made by an iron piece of same weight. By keeping in mind the above classification many normal and abnormal sounds may be quickly traced to their origin. There are some sounds so common with certain engines that they may as well be mentioned here, although, strictly speaking, they should be treated of with other abnormal noises.

The first is water in the cylinders, which causes a thump on one or both ends, and may often be detected by observing water gushing out around the indicator pipe and cocks and piston rod stuffing box. If suspected, the cylinder relief valves or drain cocks on the corresponding ends of the cylinder must be opened. If, as often happens, there is no pressure in the cylinder, the cut-off may be run in until the pressure in the receiver rises above the atmospheric pressure, unless the cylinder drain connects with the condenser.

There is another sound, a harsh, rasping one, often heard in a low-pressure cylinder when the engines are slowed down preparatory to entering port. It is caused by a small amount of water in the cylinder and will cease when the engine is speeded up again. If that noise should occur when the engines are first underway, it might properly be attributed to the piston rings being too tight. If caused by a small amount of water, it does no harm, though unpleasant to listen to.

To discover in which crank pin or main journal there is a thump accompanied by a jar of the engine, take notice which crank passes

the dead point when the sound is heard. An additional check is to flood the journals in succession with oil or water through the oil pipe and observe if the sound is deadened. First try the crank pins, then the crosshead journals, then the main shaft journals. If the sound continues, it may be that the oil or water does not get access to the bearing, or else the noise may be due to the piston being loose on the rod, or the follower loose on the piston. Thumps due to loose crank pin or crosshead brasses, loose piston, or loose follower, will be greatly reduced if nearly all the load is taken off the cylinder by readjustment of the various cut-offs. It may also be reduced by linking-up. If it is due to a loose core plug, follower bolt, or anything solid inside, being struck by the piston, the foregoing measures will have no effect in diminishing the noise and the engine must be stopped as soon as possible. When the piston springs are not tight enough, they alternately contract and expand towards the end of the stroke, causing a knocking of the ring against the walls of the cylinder, giving out a clear, metallic clacking, without producing any vibration in the engine such as would have been observed if the journal brasses, piston or follower were loose. The only damage will be leakage of steam and the remedy may be postponed until arrival in port. Metallic packing sometimes gets loose and causes a clacking sound at the end of one or both strokes which may easily be mistaken for something inside the cylinder, especially if at the same time the loose packing causes a poor vacuum, which latter gives rise to a coincident jar of the whole engine from the consequent change of distribution of the pressures on the piston. A sudden stillness prevailing in the engine room is one of the most alarming indications, for it is a sign that some large journal is getting hot. If accompanied by an odor of burning oil the indication is confirmed. If a thump is heard about the cylinder when the piston is near the middle of its stroke, it must obviously come from the slide valve, and is probably due to the valve being loose on the stem or else striking something at the end of its stroke. If the latter is the case, the noise will be stopped by linking up, which will reduce the travel and make it possible to continue working the engine, although it is advisable to stop as soon as practicable to remove the obstruction.

A sharp, clear-cut sound of exhaust steam in the exhaust pipe indicates rapid condensation and the amount of the injection may be somewhat reduced. A dull sound in steam or exhaust pipe is an indication that the boilers are foaming, or, at least, that the steam is wet.

A thump in the air pump may be caused by one or more valves being carried away. If this is the case there will be a fall of vacuum at the same time.

It is a very difficult matter to discover from which direction a sound comes, owing probably to the reflection of the sound waves as they strike different surfaces of the machinery and bulkheads, and also to the metal being a better conductor than air. An illustration of this is the case where search was made in the rigging for the cause of a sound due to vibration of the main injection valve in the bottom of the ship. The true location was discovered by observing that it did not coincide with the rolling of the ship, but was exactly coincident with the motion of the independent circulating pump.

If brass cuttings are seen to come from the pipe which permits sea water to circulate in the stern tube and into the bilge, it shows that the lignum vitæ strips have worn down to the brass; the propeller shaft is consequently out of line.

In order to be able to judge quickly and correctly the cause of any irregularity in the working of an engine it is important to have a thorough knowledge of all the details of its construction and present state of adjustment.

Routine at Sea.—Upon coming on watch, see that the speed of the engine, position of the throttle and cut-off valves are correct, and that all steam and vacuum gages, thermometers, etc., give normal indications in accordance with the table made out from previous experience.

See that all journals, slides, piston rods, valve stems, etc., are properly lubricated, water in bilge and feed tank not too high, and that the bilge strainers are clear. Each man informs his relief of all orders relating to that part of the machinery under his immediate care. The machinist and the machinist's mate should read up the logs of the two preceding watches to see if anything unusual has occurred which their predecessors may have forgotten to mention.

The machinist's mate is principally occupied in observing the movements of the engine as a whole, receiving and making reports, regulating the speed, and noting data in the log. The oilers should keep moving about that part of the machinery under their care, making systematic inspection of all bolts, nuts, keys, set-screws, stuffing-boxes, and pipe joints, to see if they require tightening, also to keep a lookout for any indication of faulty lubrication, or heating of slides or journals, applying proper remedies when possible, and, if not, reporting the fact to the machinist in charge of the watch. The oilers must also, at regular intervals, inspect the bilge strainers and gage glasses on separators and feed tank, drain the separators when necessary, and notify the water-tender if the feed tanks are too full. Examine stern tube bearing from time to time by feeling the gland and taking the temperature of the water from the cock fitted at the after bulkhead. If the bearings show any signs of warning, slack back the gland and open water service cock over the stuffing box.

The machinery on other decks, including steering engine, is often cared for by one of the engine room oilers.

An easy method of cleaning a bilge strainer is to use a jet of steam. The usual method is to put a strainer in an ash pit and let the heat melt off the grease.

The oiler must keep the hand rails and engine wiped off, ladders and platform swept, and see that everything about the engine room is at all times secured for sea, collect data for the log, and give it to the machinist's mate at the end of every hour. He must taste the water in the hot well occasionally. If found brackish, it is an indication that the hot well relief valve is stuck open, or that the condenser tubes leak.

All use of oil in the cylinders must be avoided, a small quantity of vaseline being applied to the piston rod occasionally, if no oil pipe is fitted to the stuffing box.

The number of wicks in a cup should be increased or diminished as occasion requires and sight-feed oil cups adjusted. The wicks should be taken out, dipped in oil, and replaced from time to time; those that have been clogged with impurities in the oil or soaked with water are to be renewed, taking care not to let them get twisted or extend down far enough to touch the journal. Oil which

accumulates in drip pans under journals and eccentrics must be removed before it runs over.

If forced lubrication is installed great care must be taken to keep the oil pumps in operation, the oil clean and a proper supply to the bearings; that no waste or foreign matter gets into crank pits, and that settling tanks are functioning properly. A lack of discharge from any bearing or guide must be immediately investigated and remedied. Upon entering a shallow, muddy harbor or river or upon grounding all sea suction that can be temporarily closed should be closed; close attention must be paid to injection valves and piping and every precaution taken to prevent sand or mud entering condensers and engine bearings; should this occur the condenser tubes may be partly or wholly clogged, shutting off the circulating water and lowering the efficiency of the condenser; the passages through the bearings may be clogged, causing the bearings to heat, and, if water is being sprayed over any bearing or guide, sand will be deposited on the journal or bearing surface, cutting and causing it to become heated.

Changing Speed at Sea.—When maneuvering with a fleet, keep the cut-off well run out, to insure prompt starting ahead or astern in case engines should be stopped. If required to increase the speed, open out the throttle (or run out the cut-off further) as quickly as possible without causing the index of the steam gage to tremble. Keep the water-tender informed as far as possible in advance of all proposed changes. If the change is to be permanent (to last two or more hours) notify the oilers to regulate the supply of oil. After the cut-off, pressure, etc., are materially altered take a set of indicator cards.

Coming into Port.—Upon receiving notice that the ship will come to anchor at a given time, pass the word to the water-tender, and also call down whatever part of the relief watch that is needed to bring the ship to anchor, and get the capstan, winch and steam launch engines ready. Make an inspection of all machinery in operation to see if the reports that have been made during the run of repairs required are correct and complete.

As soon as the engines are slowed, shut water off all journals and run the cut-off out if necessary for quick handling of the engine.

Arrival in Port.—Upon receiving word that the engines will no longer be required, notify the water-tender; put the engines in the position most convenient for the work that is to be done, to save trouble of using turning engines; shut the engine stop valves; open drains on cylinders and valve chests; take out wicks; empty and clean oil cups, scalding the latter with hot water and a little soda; wash out all oil ways in crank pit journals with hot water and syringe, then oil the bearings; clear all oil cups and pipes about engines; stop all oil pipes and cups with plugs of waste, if they are not otherwise covered; cover the ends of journals with gaskets and also the piston rods, pump rods, and valve stems, where they enter the stuffing boxes. Set up on the stern tube stuffing box gland. Clean the engine frame and moving parts while the grease is still hot and soft. The air and circulating pumps should be slowed down, but not stopped until the steam pressure in the fire room has fallen and is well under control. As soon as the pumps can be stopped shut the outboard valves. If the distiller is not in use, start it up to make use of the steam in the main boilers.

ENGINE AND BOILER DEFECTS—CASUALTIES.

Loss of Feed Water.—This loss is replenished by the use of the evaporators or from fresh water carried in the reserve tanks or double bottom; the amount necessary to be made up varies largely with different ships and ranges roughly from $\frac{1}{4}$ ton of water for each ton of coal burned when steaming at high powers to $\frac{1}{2}$ ton of water per ton of coal at low powers. The increase is due to the losses in the auxiliary machinery whose losses are nearly constant at all powers. Feed water losses occur through leakages from the boiler, stuffing boxes, drains, air pumps, and other fresh water pump rods, relief valves, leaky joints, tube sweeping by steam, whistle and siren, leaky valves, etc.

Surplus Feed Water.—Should there be no necessity for a make-up feed, and the water in the feed tank is increasing, an examination should be made at once. Test the feed water; if found brackish it may be due to leaky condenser tubes, or the evaporators may be priming. If the water is fresh, examine the suction valves

to the reserve fresh water tanks to see if they are partially open or leaky.

Renewing a Damaged Grate Bar.—The fire is cleared away from the neighborhood of the defective bar and the bar hauled out. The draught is checked in that furnace and a new bar placed by means of the fire tools.

Burst Gage Glass.—With the self-closing water gages, the flow of water and steam is automatically shut off and a new glass is readily inserted. Without this arrangement the cocks on the boilers must be immediately closed and the glass renewed. To prevent any bending strain on the glass with its consequent breakage, the fittings should be originally well lined up. The glass should be of the correct length with its ends square; place the packing on the glass, insert it in the stuffing boxes and screw up the glands hand tight. Warm up the glass gradually by means of the steam and drain cocks, then, with the water cock open and drain closed, the glands can be adjusted as required.

Water gages are tested by first closing the bottom cock and opening the drain cock, when steam will be blown through, showing the cock at the upper end is clear. If the upper cock now be closed and the lower opened, water will be blown through the glass and drain, thus showing both openings, top and bottom, clear. In thus testing water gages, if the water rises sluggishly in the glass to the level of the other boilers after closing the drain cock, it will indicate a partial choking of the lower passage or cock. If repeatedly blowing the upper and lower cocks does not clear the obstruction, the passage will have to be cleared by removing the screwed plugs, and passing a small rod through the passages, lowering steam and removing the glass if necessary.

The indications of the gage glasses should be verified by the test cocks. If the water in a boiler is clean, it is difficult to tell a full glass from an empty one by an inspection of the glass. In this case close the upper and lower cocks and open the drain, when the water will flow out slowly if the boiler is full. Next close the drain and open the bottom cock, keeping the top cock closed, when the water will be seen to ascend until it fills the glass.

Low Water or Water Out of Sight.—The following requirements of Naval Instructions fully cover this emergency :

“Low water in a steaming boiler is one of the most serious and most frequent emergencies that arise in a fire room, and is liable to cause distortion of the tubes or other injury to heating surfaces, serious steam and water leaks, or the explosion of the boiler. It is generally the result either of inattention on the part of the water tender, or of his attention being diverted to other duties. Failure of the feed pumps, leaks developing in the feed discharge line, hot or low water in the feed tank, a defective check valve, or water gages giving false indications due to defects of the cocks fitted to them are other causes which may lead to low water, if not discovered and corrected with sufficient promptness.

“Whenever the water in any water-tube boiler falls below the lowest try cock and out of sight in the gage glasses the fires shall be hauled, or, if burning fuel oil, the supply of oil to the burners shall be shut off, the safety valves shall be opened cautiously to relieve the pressure gradually and the feed check and boiler steam stop valves shall be closed. The dampers and the furnace and ash pit doors shall also be closed. All air leaks to the furnace around these doors shall be stopped. Under these conditions the boiler will gradually cool, and any parts that may have become overheated will be subjected to an annealing process. No attempt shall be made to restore the normal water level by increasing the supply of feed water. Fire extinguishers, if fitted, or otherwise a fire hose or wet ashes, shall be used to quench or deaden coal fires before hauling them.”

Feed Pump Inoperative.—If the feed pump is in operation and the water does not rise in the boiler, feel the beat of the check valve and see if the temperature of the feed pipe near the boiler is approximately that of the feed water; for the check valve may be jammed and not operating properly, or hot water may be flowing back to the feed pump. If check valve is jammed, tapping with a hammer frequently starts it into operation.

Examine to see that the proper suction valves are opened and that there is water in the tank. Examine for boiler leaks, such as leaky tubes. If the pump is running very rapidly, deficient supply

of water will probably be found. See that no delivery valve other than that to the boiler is open. Examine for leaks on the suction and delivery pipes and for any derangement of the suction and delivery valves of the pump.

If the pump gets hot from the water working back from the boilers, it works through the delivery valves and increases the pressure in the suction side, thus interfering with the lifting of the suction valves. The application of cold water over the pump will usually start the pump working.

If a feed pump refuses to start, there may be considerable water in the cylinders. Keep the pet-cocks open for some time, see that the valve gear is oiled, and pry it over by hand from side to side. Failing in this, try the pump on a cold boiler in which there is no pressure.

Should none of the above plans succeed in getting the pump to work, there is probably some derangement about the valve gear or auxiliary ports and passages, necessitating a removal of the bonnet and a complete examination and overhauling.

If the water barrel is badly worn, the pump may pump against a low pressure, but not be able to force water against the boiler pressure.

As soon as difficulty is experienced with a feed pump, start another on the boilers while examining for defects in the one under consideration.

Priming.—By this term is meant the passage of water with the steam from the boilers to the engines. The result of priming is sometimes low water in the boilers, and unless care is observed it may result in disablement of the engine by the cracking of cylinders and cylinder covers and injury to the pistons and rods.

The causes of priming are defective design, boilers unduly forced, bad management, and impure water.

The Design as Affecting Priming.—The design should promote proper circulation and easy escape for steam generated. These considerations are induced, (1) by sufficient area of water surface; (2) by providing for a sufficient reserve of steam by making the volume of steam space large enough to prevent fluctuation in pres-

sure; (3) by providing internal steam pipes; (4) by locating the stop valves in proximity to the direct steam.

The Effect of Bad Management.—Bad management will cause priming, (1) by carrying the water too high and thus reducing the steam space; (2) by starting the engine or increasing the speed too suddenly; (3) uneven firing or unequal forcing of the different boilers. Firing unevenly causes unequal rates of combustion in the different furnaces with unequal evaporation, thus interfering greatly with the natural circulation. (4) Dirty boilers, due to certain forms of grease, to excess of soda or impurities and to dirty or muddy water, from the boilers having not been cleaned properly. A sudden change in the character of the feed water may also induce priming.

To Check Priming.—Slow down the engines and check the rate of combustion, increase the feed to check ebullition and also to maintain the water level, which is lowered by priming. If only individual boilers of the set in use are priming, this may be regulated by partially closing their stop valves. If all are priming and the cause is the impurity of the water, the feed tanks are frequently allowed to overflow to the bilge and clean water from the reserve tanks used, together with the use of the surface blow.

After trial, if these methods fail and the boilers continue to foam at the speed set and all other conditions are good, the cause is probably an insufficient number of boilers in use, which are forced beyond their capacity. In this case the speed should be reduced or an additional boiler started.

Indications of Priming.—Priming may be recognized by the turbulent and broken condition of the column of water in the gage glass, the water often passing continuously through the glass and not remaining at a steady level.

The separators fill with water, necessitating the drains being kept well open.

The water coming over into the cylinders and valve chests, occasioning knocks or crackling sounds at the ends of the stroke. Drain cocks on the cylinders and valve chests should be kept open and the engine speed reduced.

The speed of the main engines and auxiliaries is reduced by priming, and as the steam and water passing into the condenser is increased in quantity, the air pumps become overloaded and the feed tanks filled.

Leaky Tubes.—Tubes may split due to poor manufacture, or holes may appear in them owing to pitting; they may burst owing to having been worn thin by corrosion, or pull out of the headers or sheets due to their ends being worn off or not having been properly expanded. In all cases pressure must be removed from the boiler, manhole covers removed from drums, or handhole plates from headers, and the tubes plugged or renewed. In boilers having small bent tubes it is better to plug the leaky tube and defer permanent repairs until port is reached, but in the straight tube type a leaky tube can be replaced by a new one almost as quickly as it can be plugged. In the bent tube type all the water must be removed from the drums, which must be entered, but in the straight tube type the water level is reduced just a little below those handhole plates which are to be removed. Should a tube burst or leak badly the furnace and ash-pit doors must be kept closed and the damper wide open to allow for the escape of the steam up the funnel; in cases of bad leaks it is necessary to start the blowers to protect the personnel from the escaping steam.

Hot Bearings.—Hot bearings may arise from various causes, such as the following:

1. Insufficient lubrication, that is, oil ways not sufficient or improperly placed, or to the neglect of the oiler.
2. The presence of dirt or gritty substances in the bearing. To avoid this, open the bearing and clean it from time to time when overhauling, clean out oil cups and pipes; use clean oil.
3. Improper adjustment, the amount of clearance between journal and brasses too small, or brasses not set square.
4. Bearing surface of insufficient area, the pressure between the working surfaces thus becoming so great as to prevent a film of lubricant passing between them.
5. Working surfaces not in line, causing abnormal pressures in places.

6. Bad fitting due to poor workmanship, so that the pressure is not uniformly distributed.

To Cool Hot Bearings.—If a bearing is running “warm” only, the usual remedy is to increase the supply of oil, together with the water service, or the application of a mixture of oil, soft soap and water through a syringe, finishing up with the cleaning of the oil cup and feeding with clean oil. If the temperature increases, the bearings may need slackening back a slight amount. With main bearing, thrust and similar parts, this can be done without stopping; for the crank pin, crosshead brasses, etc., the engines will have to be stopped.

When a bearing is found to be “hot,” the engine must be slowed down, and if the heating is in an advance stage, the engine must be stopped and the bearing cooled by the application of oil. Powdered black lead or sulphur mixed with the oil is often found beneficial in carrying off the heat due to the friction. It may become necessary to slack back the brasses to a slight degree, taking care that the clearance given is not so great as to produce an abnormal pounding.

The work on the engine showing the heating may also be reduced by the independent linking-up gear.

The Application of Water.—While no harm may result from the application of water to a “warm” bearing, it must be very carefully used in the case of a *hot* bearing. The sudden application of water to overheated metal may readily cause cracks or distortion due to unequal contraction. The best method to be employed in cooling bearings can be determined only by individual experience. Should water be used on the bearings during the working of the engine, it should be stopped some time before coming to anchor, and oil employed to prevent the journals rusting after the engines are stopped.

Hot Piston Rod.—If a piston or valve rod commences to heat, it is usually accompanied by the smell of burning oil, and on oiling a dense smoke will rise. If warming only slightly above the normal, the heating can often be checked by lubricating with a swab dipped in vaseline or petrolatum.

If the rod is *hot*, slack the gland, and, in order to remove the load from the engine to prevent the rod from bending, slow down and if necessary stop the engines. Apply oil carefully; water should never be played upon a *hot rod*, as the sudden cooling and contraction of one part will in all probability cause a bent rod, especially if it is hollow. With the hot rod there is also danger from a break-down, owing to melting of the soft metal used in the pistol rod stuffing boxes.

To prevent the troubles arising from a hot rod, care should be taken to see that the bushing at the bottom of the stuffing box and the gland are not too tight, causing excessive friction, that the piston rod works centrally in the stuffing box gland and bush, and the rod kept lubricated, not only by the oil pipe leading to the stuffing box, but also by application of vaseline or petrolatum by hand from time to time.

Loss of Vacuum.—The vacuum may be impaired by an insufficient amount of circulating water due to the injection strainers or ends of the condenser tubes becoming partly choked, or the circulating pump may have slowed down or stopped.

Condenser tubes made foul or dirty from grease or muddy deposits on the steam side will affect the vacuum by reducing the conductivity of the metal of the tubes. Tubes may be cleaned by introducing into the steam side of the condenser a solution of caustic soda and then boiling it by means of steam and afterwards pumping the contents overboard. Care should be taken to see that none of this mixture is introduced into the boilers.

The vacuum may fall from air leaks into the condenser from piston rod glands, low-pressure relief valves, drain cocks and fittings about the condenser left partly open. Leaks from the drain valves, glands, and pipe connections of the various auxiliaries in connection with the condenser are fruitful sources of a low vacuum, especially from idle auxiliaries.

Faulty Action of Air Pump.—The vacuum may be impaired by the defective action of the air pump, due to the following causes:

1. Defective valves or valves carried away.
2. Defective design due to large clearance space between the bucket and foot valves at the end of the stroke.

3. Poor fitting bucket or barrel worn.

4. A defective vacuum gage may give a wrong impression of the actual vacuum in the condenser, the cock may be partly closed, or the connecting pipe leaky. A comparison with another gage or indicator diagram will show if there is any defect due to the gage.

In the vertical single-acting air pump, so long as the bucket valves and either the head or foot valves are in good order the pump will work satisfactorily. If the bucket valves are impaired or carried away the effect is soon observed in destroying the vacuum.

Casualties.—The same general derangements or accidents are liable with the water-tube as with the fire-tube boilers, but owing to the small amount of water carried in the water-tube type the results due to low water or feed failure will come much more quickly and will require quick thought and action. If a tube should burst, the water will be discharged into the furnace, deadening the fires and passing out of the smoke pipe as steam. On the discovery of a ruptured tube the fires should be drenched at once and hauled, to prevent any damage to the other tubes, and that boiler cut off from connection with the others. Even with this promptness a number of tubes will be found bent and distorted, but with the possible exception of a few, these can be straightened in place without any other work.

In cases of damaged tubes in one boiler it is advisable to close the feed on that boiler and at once carefully look out for the feed of the others, to prevent a similar occurrence to others of the set.

In large tube boilers, with split tubes or a rupture in the steam or water drums the consequences may be more serious. The general principles of fire-room practice, however, will apply in these cases and the procedure will be largely governed by the nature of the accident and the good judgment of those in charge.

Owing to the small quantity of water contained and the great elasticity of the water-tube boiler, steam can be raised quickly. If no particular haste is required, the average time is about two hours. If it is necessary, small tube boilers can obtain steam in thirty minutes, large tube boilers somewhat longer, but in general, steam can be ready by the time the engines are warmed up.

CHAPTER XVII.

PORT SERVICE, EXAMINATIONS, ADJUSTMENTS AND REPAIR.

When cool remove the manhole plates from the cylinders, and covers or sight-hole plates from valve chests. Inspect the interior for marks of cutting, see if follower bolts, piston nuts are tight, etc.

If lagging about cylinder heads, cylinders or valve chests shows much moisture when the engines are in operation the lagging should be removed and search made for the cause. This may be due to cracks or holes in the metal, or leaks about pipe or valve joints. If due to small cracks in the metal they can often be repaired by fitting sheet metal patches secured with bolts. If holes are found, not greater in diameter than the thickness of casting, they can be repaired by drilling the hole, tapping it, and fitting therein a metal plug.

To examine the piston remove the follower bolts, follower springs and packing rings. In testing the spring of the latter it should have sufficient elasticity to sustain its own weight when placed in any part of the cylinder, and with sufficient tightness that it can be pushed down by hand. Examine springs and test their elasticity and set out or renew those requiring it. The condition of pistons, rings, springs, etc., found on examination must be noted in the steam log.

The interior of a cylinder should have a good polish all over. If there is a dark streak in the direction of its axis, it indicates that the springs are not tight at that part of the piston. The tightness of a piston may be tested by closing one end of the cylinder, putting the link in mid-position and working the engine with the turning engine. A candle on the end of a stick being moved around the circumference of the piston will show by the flickering flame whether much air leaks past as the piston approaches the closed end of the cylinder. If the cylinder is closed at both ends, the desired infor-

mation is obtained by observing the amount of air pumped in and out the indicator pipes, or as described on page 235, Chapter X.

After inspection, oil the interior of the cylinder and wearing surface of valve chests with petrolatum or vaseline and replace the plates, taking notice that the openings of the relief valves are not choked. Especial care should be taken to see that all nuts and fastenings are properly secured and that no tools, waste, etc., are left within.

Proceed with the repairs found necessary during the run, beginning with the most important and arranging the work so that the different gangs of men will not interfere with each other; that steam will not be shut off the auxiliary pipe by one gang, just when it is wanted by another; that floor plates will not be taken up in a passageway where there will be much passing, and many other like considerations. Repack all stuffing boxes needing it.

In case there are no repairs, proceed with the routine inspections of various parts of the machinery as required by Naval Instructions under "Care and Preservation of Boilers and Machinery." Also clean out the feed tank and scale evaporators when necessary. Leaky condenser tubes are located by filling the steam space with water after having removed the bonnets over the ends of the tubes. Grease may be removed from the condenser tubes by filling the steam space with a solution of soda in fresh water and boiling by admitting a jet of steam at the bottom.

It is customary to devote the morning watch to cleaning bright work, working all levers, valves, cut-off gear, etc., about the department. Iron and steel are sometimes cleaned with emery cloth and oil, taking care to rub in the direction in which the piece was finished by the manufacturer. The use of emery cloth is objectionable, as particles of emery may find their way to the bearings with injurious results; bath brick and oil or fine crocus cloth is safer, but even with this great care should be used to avoid the effects above mentioned. After polishing, such pieces are wiped with an oily rag to leave a very thin film of oil on the surface. Soiled spots on the paint work are removed by washing with soap and water or with turpentine on a rag. Parts of the engine frame and bulkhead where the paint is blistered or peeled are scraped, washed with lye or turpentine and

repainted. Bulkheads and other paint work are scrubbed with soap and water. Ladders and floor plates are scraped, washed with lye and dried. Hot, dry ashes will remove grease from floor plates. Wooden lagging is scraped and rubbed off with boiled linseed oil and a rag.

The main engines must be oiled and moved at least one turn of the crank every day, with the links in full gear at the time so that the main valves will be moved through their full travel.

Auxiliary engines must be turned daily; and run by steam as often as found necessary by experience, not less than weekly. Those on the upper deck must be kept well drained and covered up in cold weather, if there is danger of freezing. The engine room bilges must be cleaned within a day or two after reaching port, and once a week after that. Oil can be readily removed from the bilges by using scalding water from the boilers if this can be spared. If cold water is used, caustic potash, or better, soda, must be mixed with it to form a strong lye, which is applied with a swab. All thick deposits must first be scraped up.

While in port a fair share of time should be devoted to keeping spare parts of machinery in good condition.

Double Bottoms.—The last work will be cleaning and drying out the double bottoms and renewing the cement and paint as required. Before entering a compartment, let down an open light. If it does not burn brightly, the compartment must be ventilated; this is easily done after the manhole plate is removed, by starting the compartment pump as if to pump out water. If, as sometimes happens, there are no pipes for pumping out the compartments, a small blower should be procured and provided with a flexible hose to discharge air into the compartments.

REPAIRS AND ADJUSTMENTS.

While it would not be possible within the limits of a general and elementary text-book to enter into specific methods of conducting repair work on board ship, a few hints are given in the following pages regarding the more common operations met with in the adjustments and repairs found necessary in the working of marine machinery.

Adjustment of Main Bearings—Crank Pin and Crosshead Brasses.

—The necessity for an adjustment of a bearing will be determined by the manner in which it has been working, both as regards noise and temperature, and the operation is one requiring skilful and attentive work. It is essential that the bearing should be kept properly adjusted so as to produce neither hammering nor heating. It is impossible to give a fixed rule for the amount of clearance between the journal and bearing of fast revolving or oscillating bearing surfaces; this is usually determined by experience and judgment, and is governed by the size of the journal. In ordinary cases the clearance may be made about .002 of the diameter, and modified by individual experience with the particular machine.

To adjust the amount of clearance in a bearing, remove the cap and top brass and liners or distance pieces. Lay a piece of soft lead wire about $\frac{1}{16}$ inch diameter along the journal parallel to the axis, or three pieces placed at equal distances apart on the journal and at right angles to the axis. Replace the distance pieces, brass and cap, and screw down hard and evenly on the nuts. The lead wire is squeezed out flat. The position of the nuts is carefully observed and marked, the cap and brass removed, and the thickness of the flattened lead wire measured by a micrometer gage gives the clearance along the bearing, which should be of uniform amount. This operation is called *taking a lead*. Several leads may be taken in order to set the clearance at the exact amount required. After removing the leads the distance pieces and thin liners should be carefully fitted so that when the brass and cap are replaced, and the nut screwed up against the distance pieces, the nuts are in the position marked for the required clearance. Before putting a bearing together, carefully clean and oil the journal and brasses. When a crank pin bearing is readjusted, insert a pinch bar between the brasses and a fixed part, such as crank arm, and move the bearing along the pin in several positions, to see that the bearing is free and does not bind. The leads should be kept and labeled, so that when compared with one subsequently taken at the same marks, an indication of the nature and amount of the wear is given. The officer in charge of the engineer department should keep a record of all adjustments.

When time is a matter of consideration, and there is a knock

requiring removal, the liners may be removed, and the nuts screwed hard home; note the distance they have moved, and then slack them back to half this distance, or by an amount determined by experience with the particular circumstances of the case.

The operation of adjustment can be facilitated by marking permanently with lines and numbers the faces of all large nuts used in making adjustments, together with a reference line on the adjacent metal from which measurements may be taken. This will also prevent confusion and the disfiguring of finished parts by numerous markings from center punch, chisel, etc.

Refitting a Bearing.—When bearings become hot the solid brasses are frequently cut and the journals scored, in which case a certain amount of refitting is necessary. On board ship this refitting does not extend beyond removing or smoothing down the rough spots. The top and bottom brasses are removed, the oil ways chipped out and the journal calipered to see to what extent it is out of true. If found rough or scored, it is filed evenly, care being taken to preserve the cylindrical form, true with the axis. This is followed by a careful dressing with the oil stone. The brasses are filed and scraped until a good bearing is obtained along the crown and all flaky material removed. Since a bearing at the sides serves no useful purpose, these parts are given a good clearance. Each brass is revolved or lightly rotated on the journal by hand in order to test the contact between journal and bearing; the journal having been lightly smeared with a little red lead, the high spots on the bearing will be indicated by the adhering red lead and must be scraped down until a satisfactory bearing is obtained. If time and facilities are at hand, a more satisfactory process is to test the contact with both brasses in place and secured.

If brasses lined with white metal are cut or worn sufficiently they must be relined.

To Reline a Bearing with White Metal.—The bearing is heated in a clean coke fire and all the old metal melted out. The bearing is then cleaned with acid and the inside surface tinned. A hollow mandrel of cast iron of the same size as the journal is properly supported in place to obtain the necessary thickness of white metal. The ends of the bearings are closed by thin plates and clay, and the

white metal is poured into the annular space from a melting pot or ladle. On cooling, the white metal is well pene- into the recesses, then bored and scraped to a proper fit on the journal.

Other Adjustments.—Crosshead slippers require frequent adjustment. For examination the slippers are removed and scraped as desired. Between the slipper and crosshead are placed liners with which the clearance between the crosshead guide and slipper may be regulated, and the piston rod adjusted to move in a central line through the axis of the hole for the piston rod stuffing box.

The link gear, owing to the wear, will require frequent attention and adjustment. In the adjustment of eccentrics, the degree to which the clearance between the bearing surfaces is reduced is best tested by disconnecting the rods from the link and moving them by hand, but leads should also be taken.

Thrust Bearings.—To effect adjustments due to the wear on the thrust collars, the collars are moved back by means of the adjusting nut provided in the case of horseshoe collars, or, with the ordinary fixed collar type, the whole bearing is moved slightly by means of adjusting screws provided on the bearing.

Shaft coupling bolts should be withdrawn and cleaned from time to time to prevent seizing. This may be taken advantage of to see if the shaft is in line by measuring the opening between the two couplings after they are disconnected. If the shaft is in line the faces of the couplings will be parallel to each other, the clearance being the same all around.

To Straighten a Bent Piston Rod.—If the rod is large or the bend is short, the rod is heated to a dull red with a wood fire (to prevent scaling). The rod is then placed in a lathe and a strong-back employed bearing on three points, the outside bearings being underneath the rod, and the middle one above, midway between the other two, and at the middle of the bend. By means of jacks, pressure is applied at the middle bearing and the rod forced back into a straight line. Owing to the form of the strong-back and points of contact no strain other than the weight of the rod is brought on the lathe centers. To prevent injury to the surface of the rod at the point of contact, these parts of the rod should be protected by copper pieces inserted between rod and bearing point.

If necessary a light finishing cut will be taken on the rod after straightening. It is hard to perfectly straighten a bent rod.

To Test the Condenser.—The tubes of a condenser are liable to split; to become pitted from corrosion with a formation of pin-holes; the tubes may leak at the packed ends, or at the screwed stays of the tube sheet. To test and locate the leakage, close tightly the main injection and outboard discharge valves. Close all valves and gages. Remove the condenser bonnets or manhole plates. Blank flange the exhaust pipe from the low-pressure cylinder and the suction pipe to the air pump. The steam side or space outside the tubes is filled up with water and a pressure of from 15 to 20 lbs. per square inch is put on the condenser. The tube sheets having been wiped dry, defective tubes will be indicated by water running out the ends. Poorly packed tubes or leaky stay bolts will show leakages around the individual parts. Leaky tube ends are usually due to slack glands, and all can be reached through the manholes in the heads. For leaky tubes not directly in line with the manholes, and when it is desired not to plug but to renew them, the condenser heads must be taken down, the glands and packing removed and the defective tubes pulled out and renewed. If the condenser head is removed, the pipe opening from the injection and discharge should be blank flanged as an additional safeguard against the entrance of water into the ship.

In examining the condenser the steam side should also be inspected and the condition of the tubes noted with reference to grease. About once in six months a tube should be drawn and the condition of the metal examined.

Copper Pipes.—Cracks or perforations in a copper pipe are generally repaired by brazing a patch over the hole or split. Clean the metal carefully with file, emery cloth and acid. A patch of the same thickness as the pipe is cut out of sheet copper, shaped to the pipe and wired in place. The pipe is then heated in the forge with charcoal or coke fire. Spelter or hard solder mixed with borax as a flux is placed on the defective part inside the pipe and the whole heated, the melted spelter being manipulated by a poker to keep it from running directly through, and causing it to run in between the patch and the surface of the pipe, thus forming a joint between the two.

By the application of loam to the pipe, the heat may be directed to the part under repair and to protect the neighboring parts from injury. Care and skill must be exercised to prevent burning the copper.

A small split in a pipe may be temporarily repaired by wrapping with heavy canvas and red lead and tightly wound with marline or copper wire. A better plan would be to shape a piece of copper to the affected part and firmly secure it in place by two or more bolted clamps, with a layer of light, soft packing between the patch and pipe.

A small hole in a thick feed pipe may frequently be stopped by tapping out and riveting in a small screwed plug.

Small pin holes may be temporarily closed up by the use of soft solder.

Leaky Joints.—All leaky joints noted or developed while running should be promptly set up on or remade, owing to the corrosive effect on the neighboring parts, together with the probability of the progressive increase in the leak and the waste of fresh water. The flanges of flanged joints must be sufficiently thick, and the pitch of the securing bolts small enough to prevent the flanges springing when screwed up, or when pressure is applied. The two flanges must be parallel to each other, otherwise, if they have to be sprung together by the bolts, it is almost impossible to maintain a tight joint. Also, the flanges should not be so close to a bulkhead or obstruction that the nuts cannot be properly tightened up.

Spare Parts.—To provide for emergency and accident as well as for the ordinary wear in service, spare parts are usually carried. In the naval service the parts carried are usually greater than in mercantile vessels, to make the vessel more self-supporting on distant service, or, for military reasons, to avoid the necessity for the ship leaving her station for work which would require a repair shop. For the same reasons a small machine shop is equipped. A list of spare parts carried, together with the equipment of the machine shop on board a modern armored cruiser or battleship, is given in the appendix.

Dry Docking.—Before entering dry dock see that the ship is brought to an even keel by the adjustment of engine and fire room weights and the disposition of the water and coal carried. While in dock especial care should be taken that the arrangement of weights is not materially disturbed, that the coal and stores are equally used on both sides, and that boilers, if emptied or filled, are worked in pairs, so that the even keel is maintained.

On entering dock, as soon as the engines are required no longer, stop the circulating pumps and turn the exhaust from the auxiliary system into the atmosphere. Connect up a fire hose from a hydrant on shore to the feed tanks, and during the stay in dock the water required for the boiler and for the auxiliary machinery will be taken from this source or from reserve tanks.

Fire hose is led out each night, connection being made to the fire main from a hydrant on the dock.

The work of examination, repairs, adjustments, etc., having been prepared for in advance, and the engineer's force having been divided up into groups for the work in hand, the following is carried out: The inspection, overhauling and repacking of all sea valves and clearing their strainers, renewing zinc protectors, etc., taking care not to remove more valves in a day than can be replaced before night, as required by the Regulations. Also inspect the interior of the pipes as far as possible while the sea valves are removed, and note any pitting or deterioration of the pipes.

The officer in charge of the engineer's department is required to look after all inlet and delivery valves comprised within the limits of that department, as the following:

Injection valves for main condensers.

Injection valves for auxiliary condensers.

Outboard discharge valves from main condensers.

Outboard delivery valves from auxiliary condensers.

Sea injection valves from engine and boiler compartments for auxiliary pumps.

Overboard discharge valves in boiler compartments from auxiliary pumps.

Overboard discharge valves for bottom and surface blow.

Examine the propeller shafting, parts not covered with brass to be scraped, cleaned and painted. Examine the zinc plates on the stern tubes and renew if necessary. Inspect the propellers and see that all bolts are tight. Get an accurate measure of the space between the top of the shaft and the top of the stern bearing to determine how much the bearings have worn down and record it in the log. The following should be done if time permits: Remove the shaft coupling bolts of the coupling nearest the engine and the packing from stern tube stuffing box, attach a lever to a blade of the propeller, then suspend weights on the end until the shaft turns. Make a record of the weights and length of lever. By comparing such records any increase of friction due to the shaft being out of line or other causes will be discovered. The packing in the stern tube stuffing box must always be inspected by removing a few turns and judging from them whether the whole amount needs renewal.

CHAPTER XVIII.

ENGINEERING PROGRESS IN THE U. S. NAVY.

BY CAPTAIN C. W. DYSON, U. S. NAVY.

NOTE.—The following paper, published in the *Transactions of the Society of Naval Architects and Marine Engineers*, puts in most excellent form the present day status of reciprocating engines. It is highly improbable that much further increase in economy of the reciprocating engine will be made, unless it be by adopting higher steam pressure. The difficulties in the way of superior economy of the turbine at all speeds are rapidly being overcome.

H. O. S.

In preparing an article under the heading of "Engineering Progress in the U. S. Navy," the ground to be covered is of such large extent that, in the greater part, nothing but the merest notice of improvements can be given, and only the most important points of progress will be detailed. The first and most important point of all, as reacting upon the general efficiency of the navy as a reliable and economical fighting force, is the—

CHOICE OF PROPELLING MACHINERY FOR HEAVY VESSELS OF MODERATE SPEED.

In the selection of the type of machinery to be used in the above class of vessels, the following points must be taken into consideration:

(a) General character of the service which the vessel will be called upon to perform; whether she must keep the sea for long periods, cruising at speeds very much lower than her maximum speeds, or whether she will be called upon for very little slow cruising, but shall be held in readiness for dashes at high speed from a base to any threatened point.

(b) Greatest economy realized at the conditions under which she will be called upon to operate. This point is important, not only

from the point of financial saving in reduced fuel cost, but in the greater ease of fuel supply due to the decreased demands.

(c) Fuel capacity entailed by the demands of the service to which the vessel may be subjected.

(d) Ease of up-keep of the machinery, and degree to which the vessel, so far as machinery repairs are concerned, can be made self-supporting.

(e) Reliability of machinery when driven at high powers.

(f) Minimum weight and space required for the propelling machinery.

(g) Efficient propellers for maneuvering.

(h) Minimum of vibration of hull due to machinery in operation.

(i) Effect of vertical position of center of gravity of the machinery upon the time of roll of the vessel, in fixing the quality of the vessel as a gun platform.

The question of costs of the different types of machinery will not be considered in comparing the relative advantages of the types.

Further, the relative values of turbine reduction gear, electric propulsion and internal combustion engines for propulsion will not be dealt with, for the following reasons: The turbine reduction gear and electric propulsion are under trial in the naval service at the present time, the reduction gear being actually afloat while the vessel fitted with electric propulsion is building.

The results obtained up to date with the reduction gear have been disappointing so far as the expected economy is concerned, the results being vitiated by faulty turbines and too high a number of revolutions of propeller, 135 per minute, for the type of vessel and the speed, 14 knots. The reduction gears have, however, stood up to the work well and show practically no evidences of wear. Results are encouraging, and a great improvement is expected when contemplated changes in the turbines have been made.

Electric propulsion not having been tried out in actual service, it is considered preferable to content ourselves with the mere statement that shop tests of one of the units have been very gratifying and promise a successful end to the experiment, so far as economy of propulsion only is considered.

As to the question of propulsion by internal combustion engines, where large powers are required, there appear still to be many important problems requiring solution before units of sufficiently high powers for the purpose desired can be built. The supplanting of the steam engine, both reciprocating and turbine, for important high-power installations does not appear to be imminent in the immediate future.

Eliminating these three latter methods of propelling naval vessels restricts the choice of machinery for this purpose to the three following methods:

1. By means of reciprocating engines.
2. By means of steam turbines, impulse, reaction, or a combination of the two.
3. By means of various combinations of reciprocating engines with turbines.

COMPARATIVE SUITABILITY OF EACH OF THE ABOVE METHODS FOR NAVAL PURPOSES.

To assist in reaching a decision as to which of the three methods of propulsion best meets the requirements lettered from (a) to (i) a comparison of the performances of the dreadnoughts Delaware, North Dakota, Utah and Florida can be made; these performances include those on preliminary acceptance trials and those in actual service.

On Plate 1 (Fig. 245) are given the indicated horse-power speed curve of the Delaware and the shaft horse-power speed curves of the North Dakota, Utah and Florida.

In comparing these curves it should be noted that, due to differences in displacements and in appendages, the effective horse-power required to give the Delaware a speed of 21 knots will give the Florida and the Utah 20.82 knots, and the North Dakota 21.1 knots; at 12 knots for the Delaware, the equivalent speed of the Florida and the Utah is 12.17 knots and of the North Dakota 12.1 knots.

Assuming that shaft horse-power is approximately 92 per cent of indicated horse-power in large, well-designed reciprocating engines,

and reducing the indicated horse-power of the Delaware to shaft horse-power, the following table of comparison results:

	Delaware.	North Dakota.	Utah.
Speeds of equal E. H. P's	21	21.1	20.82
I. H. P. for this speed.....	23,400		
S. H. P., for these speeds.....	21,528	27,150	25,000
Propulsive coefficient equals (per cent) E. H. P. $\div$ S. H. P.	71.76+	56.8+	61.8+
Cruising speeds of equal E. H. P.	12	12.1	12.17
I. H. P. for this speed.....	3,600		
S. H. P. for these speeds..	3,312	3,825	3,975
Propulsive coefficient equals (per cent) E. H. P. $\div$ S. H. P.	74+	64.1+	61.6+

While the Florida curve is shown on Plate 1 (Fig. 245), that vessel is not included in the above table, as the weather conditions during her standardization trials were unfavorable and the efficiency of the propellers fell off greatly, due to these adverse conditions. This excessive falling off in efficiency of propulsion under adverse conditions of wind, sea and of ship's bottom are characteristics of turbine-driven vessels, and affect requirements (a), (b) and (c) to a large degree.

Formerly, the favorite method of comparing the relative economies of propulsion of reciprocating engines and turbines was by comparing the water per shaft horse-power of the turbines with the water per indicated horse-power of the reciprocator, in this comparison all mention of the large difference between the indicated horse-power required in the one case and the shaft horse-power in the other being carefully neglected.

The method of comparison in use to-day is the commercial one of "pounds of fuel per knot" at different speeds, and it is on this basis that the curves on Plate 2 (Fig. 246) are constructed.

CONDITIONS OF PLATE 2 (FIG. 246) CURVES.

Curves A, B and C.—Actual trial conditions.

Curve D.—Using same feed temperature conditions as for Curve C.

Curve A.—Feed heaters not in use.

Curve B.—Feed heaters in use.

Curve C.—Feed heaters in use. Feed temperature much higher than A and moderately higher than B.

As practically the same amounts of heating medium for the feed heaters were available in each case, it is justifiable to correct the British thermal units absorbed by the boilers in A and B by the amount of decrease due to the higher feed temperature of C.

Boilers in all cases were of the same type, but for A and B they were fitted with superheaters.

Boilers being of the same type, they can be assumed of the same efficiency; therefore the actual efficiency may be neglected and a boiler efficiency of 100 per cent be used for comparison of performances without sensible error in the comparison.

Curve B is not exactly comparable with curves A and C, as it does not include full expenditures for all purposes as do the others, and hence it has not been reduced to equal feed conditions with C.

In preparing A, B, C and D the following formula has been used:

$$\text{Pounds coal per knot per hour} = \frac{[T - (t - 32)] \times W}{S \times 14,500},$$

where

T = Total heat in steam at boiler pressure and superheater.

$t - 32$ = Feed temperature above 32° F.

W = Total water evaporated per hour.

S = Speed of ship in knots per hour for which W pounds of water are evaporated.

14,500 = Thermal value of one pound of coal.

On this plate is also shown a curve of excess fuel for the Utah over that required for the Delaware, all the curves being drawn with effective horse-powers as abscissae.

For the effective horse-power value required for 12 knots speed of the Delaware, the Utah requires, by these curves, 22 per cent more fuel. From the steam logs of the two vessels the average excess fuel consumption of the Utah over that of the Delaware at 12 knots is 20 per cent, thus corroborating the results obtained from Plate 2.

At the effective horse-power value required for 21 knots by the Delaware, the Utah requires 1.015 times as much fuel as the Delaware, a difference so light as to be negligible.

From the above results, it appears justifiable to decide as follows:

Should the duties of a vessel be such that she be required to steam for long periods and long distances at speeds much lower than her designed maximum speed, a less fuel expenditure per day will be required, and consequently a greater cruising radius will be obtained and less frequent recoaling necessitated should reciprocating engines be fitted rather than turbines for propelling purposes.

Should, however, the vessel operate from a fixed base, only doing sufficient cruising to insure that the machinery is kept in efficient condition in readiness for forced runs to any threatened point, the value of fuel economy at low speeds becomes minimized and, where the maximum speed of the vessel does not exceed 21 to 22 knots, either turbines or reciprocating engines may be used, the choice being dependent upon other factors than economies, which are practically equal at these speeds.

In other words, for the conditions (b) and (c), under which the American battleship fleet operates, the reciprocating engine is preferable to the turbine as a propelling engine at the present stage of turbine development.

The Navy Department is, however, thoroughly alive to the advantages to be gained by adopting rotary in place of reciprocating motion in the main propelling machinery of the heavy vessels of the fleet, and, while recognizing the present advantages held by the reciprocating engine in the matter of economy at low fractions of designed power, holds itself ready to discard the reciprocating engine as soon as the turbine designers can demonstrate by actual performance that their claims as to equality of economy at low powers with the older machine have been realized. It was with this object in view that the Department decided to install impulse turbines in the Nevada, and not because the engineers of the Department were "wobbling," as has been charged.

Condition (d)—Ease of Up-Keep of Machinery.—The claim is frequently made by the turbine advocates that while the reciprocating engine, when new, is undoubtedly more economical than the turbine at small fractions of designed power, this advantage is soon lost in active service, due to excessive wear of piston and valve rings causing large losses through heavy leakage of steam. The turbines,

not being subject to such frictional wear, would, on the other hand, retain their original economy indefinitely.

Practical experience with both types of engine in actual service comes very far from justifying this conclusion. In fact, with intelligent supervision, the reciprocating engine, particularly since forced lubrication has been applied, holds its superiority continuously.

When reciprocating-engine vessels visit the navy yards for their regular overhaul, the work to be done on the main engines is practically nil, as the machine shops and foundries of the battleships are of ample capacity to take care of all repairs that may be necessary, except such as the fitting of a new cylinder or the repair of a fractured bed-plate. The above remarks apply only, however, to ships fitted with forced lubrication, where the wear of bearings and journals has been practically eliminated.

In a letter from the senior engineer officer of the Delaware the following statements occur:

Jan. 26, 1912. Examined on each engine one main journal, one cross-head, and one crank-pin brass and found all in good condition. On passage from New York to Guantanamo Bay during a heavy gale a quantity of sea water entered the crank pits, but no trouble was experienced, although we ran at 19 knots, lubricating with a mixture of 40 per cent oil and 60 per cent sea water.

Report B. Recently leads have been taken off several cross-heads after about 30,000 miles steaming without inspection or adjustment, and they have been found in excellent condition with practically no wear.

Between July 1, 1910, and July 10, 1912, the Delaware has steamed 54,627 miles (knots) without a hot bearing.

About the only wear on the brasses on the forced-lubricating system is on the cross-heads first; then guides and then crank pins. The wearing away of the piston rings apparently starts all the other wear.

That is, keep the pistons in line and the wear on the running parts is practically eliminated.

These experiences of the Delaware are not unique, but are corroborated by the experiences of every reciprocating-engined vessel of the fleet to which forced lubrication has been applied. This system, not only insuring efficient lubrication, but also reducing wear on all running parts, decreases the initial friction of the engines and re-

duces to a minimum all shock on crank pins, cross-heads and main bearings, thus preventing hammering out of the white metal and all wearing down out of line.

When we turn to the turbine engines, however, the case is quite the opposite. Fully 99 per cent of the troubles that occur with this type of engine are internal troubles, and consist of erosion of blades and nozzles, stripping of blading, heavy corrosion of rotors, diaphragms and turbine wheels, causing destruction of balance. All of these troubles require a perfectly smooth haven in which to make repairs; and the majority of them require dock-yard facilities.

In the cases of the main engines of the three scouts, Birmingham, Salem and Chester, the Birmingham, with reciprocating engines, has always been ready for service, while her two sisters have been repeatedly laid up at the yards for overhaul of the main turbines.

Evidence of experience leads to the conclusion that a battleship fitted with reciprocating engines for propelling purposes is much less apt to be forced off her station by necessary repairs to her engines than is one fitted with turbine engines.

Condition (e) Reliability of Machinery when Driven at High Powers.—From the natures of the two machines, it would appear to be safe to decide this condition as being distinctly in favor of the turbines, as this type of engine is completely free from all reciprocating parts held together by bolts and nuts.

Experience with the Delaware's engines, however, lead to the conclusion that where proper care is taken to lock all nuts securely, and to effectively protect the engines against the shocks of reversal of direction of motion, the reciprocating engine can, even here, be regarded as nearly on a par with the turbine in reliability.

The full-power twenty-four-hour run of the Delaware, made without preparation immediately after her arrival home from Chili, demonstrates this reliability of the present type of battleship engines very thoroughly. As stated, without any preliminary preparation of engines or machinery, the vessel put to sea, and upon getting well clear of the land a full-power run of four hours was started, during which time the vessel averaged 21.86 knots per hour. Without intermission the vessel continued on for twenty hours longer, averaging for the full twenty-four hours a speed of 21.3 knots, the ship

automatically slowing down as the fires became dirty and the personnel fatigued.

Upon the completion of the trial a radiogram was received from the commanding officer of the vessel reporting that not the slightest disarrangement had occurred to either the main engines or the auxiliary machinery, and that she was ready for immediate service.

Condition for Minimum Weight and Space Required for the Propelling Machinery.—As already shown, the total heat units required to be absorbed by the boilers, both for Parsons turbines and for reciprocating engines, with battleships of the speed and power that now exist, is practically the same in both cases at full power. This indicates that, for existing conditions, nothing can be saved in the boiler-room weights or space by adopting turbines, as the same boiler power is required in the two cases.

In the engine rooms, for these powers, however, the reciprocating engine has a decided advantage in both weight and space required.

Thus, in the Delaware, North Dakota and Utah, the engine-room weights and space required are as follows:

	Delaware	North Dakota.	Utah.
Engine-room weights, dry tons....	728.26	731.23	864.69
Engine-room weights, wet tons....	773.26	785.93	919.80
Engine-rooms, length	44 ft.	44 ft.	60 ft.
Engine-room total width.....	50.5 ft.	50.5 ft.	51 ft.
Engine-room (sq. ft.) floor space..	2,222	2,222	3,060

While the turbines of the North Dakota appear to be about on an equality with the reciprocating engines of the Delaware in the matters of weight and space, these turbines were extremely uneconomical. Modern turbines of this type would require an engine room more nearly equal in length to that of the Utah, and the engine-room weights would be considerably increased.

While the reciprocating engine has a decided advantage in the features of weight and space required, under present conditions, these advantages would disappear should the necessary power to be developed be increased considerably above what is now asked for, and the advantage would rest with the turbine. Should such an increase of power be called for in future designs, or should the

ordinary cruising speed be made considerably higher than now used, the Navy Department would undoubtedly abandon the reciprocating engine and adopt one of its rotary rivals for the propulsion of its capital ships.

Condition (g)—Efficient Propellers for Maneuvering.—In considering this condition, the relation of the backing power of the vessel as compared with the maximum full-power ahead, and the time required from full-speed ahead until the vessel is dead in the water, will be taken as a comparative measure of this condition.

On Plate 13 (Fig. 255) are shown curves of indicated horse-power for backing for the Delaware and of shaft horse-power for backing for the Salem, the abscissae values being indicated horse-power and shaft horse-power in the ahead motion. At the low abscissae values, where more boiler power is always available than is necessary for the actual ahead speed being used, the backing power exceeds very considerably the ahead power in use, as, when backing for short periods, the throttles can be opened wide and the boiler power available be made use of.

When all boilers are in use, which in the case of the Delaware occurs at 25,000 indicated horse-power for the main engine in the ahead motion, and for the Salem at 14,000 shaft horse-power for the main engine in the ahead motion, the maximum backing powers can be obtained. In the case of the Delaware this maximum backing power amounts to 89.2 per cent of the ahead power, while in the turbine vessel Salem it amounts to only 41.9 per cent. That is, at these points, the backing power of the Delaware is 2.13 times as great as that of the Salem (both being expressed as fractions of the the ahead power).

At the maximum powers developed by the engines of the two vessels, the ratio of the percentage backing powers becomes: Delaware=2.27 Salem.

These results are further corroborated by the backing tests of the Delaware and the Utah upon their preliminary acceptance trials, where, with the Delaware going ahead at 21 knots and the Utah at 20, the times taken to bring the vessels dead in the water were, for the Delaware, 1 minute 52 seconds; Utah, 4 minutes 44 seconds.

	<i>Delaware.</i>	<i>Utah.</i>
Backing power divided by ahead power.....	87.5%	35.7%

These results are still further corroborated by the destroyers. These vessels can easily steam ahead at 16 knots under one boiler, but when called upon to maneuver they invariably, as a matter of safety, start a second boiler.

Conditions (h) and (i)—Minimum Vibration of Hull; Steadiness of Hull as a Gun Platform as Affected by Machinery.—In judging these points it seems only fair to base the decision upon the results of target practice of the vessels in service. If this is done, the decision could be given to the reciprocating type of machinery, as the Delaware has just won the championship of the battleship fleet, with the Colorado, another reciprocating-engine vessel, standing second on the list. From these results it appears reasonable to state that, with well-balanced reciprocating engines, no ill effects on gun fire should be expected.

CONCLUSIONS.

Basing the choice between reciprocating engines and turbines for battleship propulsion under existing conditions of speed and power upon the above comparison of relative advantages of the two types, the advantage appears to rest most decidedly with the reciprocating engines, and the Navy Department has ruled accordingly.

COMBINATION SYSTEMS.

In the search for economy of propulsion through a wide range of speeds, various combinations of reciprocating engines and turbines have been proposed, both by the Bureau of Steam Engineering and by the shipbuilders, but only one of the systems has as yet been authorized, and that one is for destroyers. It has not yet been tried out in service, but preliminary shop tests show a good gain in economy of the main propelling engines at cruising speeds. This system, as applied to the destroyers, depends entirely for its gain upon the greater efficiency of the reciprocating engine at the higher steam pressures over the efficiency of high-pressure turbines of the reaction and the high-pressure nozzles of the impulse type of turbines, no advantage being gained from increased efficiency of propellers, as the reciprocating engines are on the same shafts as the

turbines. From some points of view this combination is undesirable, and the gain in service must be considerable to justify its retention.

With the other combination systems proposed calculations indicate that if the propulsive efficiency counted upon can be obtained, these systems will all be very much more efficient than either a straight turbine or straight reciprocating engine drive at maximum power, will hold a big advantage over the straight turbine drive through all ranges of powers, and will hold its advantage over the straight reciprocating engine drive until a minimum speed of about 11 knots is reached, when the efficiencies become equal.

The "if" exists, however, and is caused by the danger of the currents thrown to the rear by the big reciprocating-engine screws seriously affecting the rate of feed and direction of flow of water to the turbine propellers. In addition, there may possibly be another source of loss due to heavy leakage of steam through the large change valves which must be fitted to control the paths of flow of the exhaust steam from the reciprocating engines.

In all of these systems, to adapt them to naval requirements, it is necessary to exhaust from the low-pressure cylinders of the reciprocating engines at a pressure of not less than 25 pounds absolute, when these engines are operating at full power, and to by-pass as few of the stages of the turbine as possible in order to obtain an increased economy of propulsion through a large range of powers.

TURBINE CHANGES TO PRODUCE INCREASED ECONOMY.

The Parsons turbine as it exists in our vessels to-day is, with very few exceptions, the same as the turbines of this type which were fitted in the initial turbine vessel, the Chester. The only improvements which have been made consist of changes in blade angles, particularly in the low-pressure stages, an increase in the number of rows of blades in these same stages, and the fitting of nozzles for the admission of auxiliary exhaust steam at several different locations along the steam path.

With the impulse turbine, however, the advance over the original naval turbines of this type, those of the scout Salem, has been rapid. The number of stages has been very much increased, both in battleship and in destroyer turbines, a drum construction has been adopted

for the lower-pressure stages, steam balance for propeller thrusts has been provided, cruising nozzles for low fractional powers have been fitted, and nozzles for utilization of auxiliary exhaust are now supplied as in the Parsons turbines.

That these changes in turbines of the impulse type have been accompanied by increase in economy has been thoroughly demonstrated by experience with the machinery of the destroyers, the economy of the impulse turbine showing up nearly, if not fully, as good as that of the reaction type. No opportunity has as yet been offered to obtain a measure of this economy increase with the battleship types of impulse turbine, nor will such opportunity occur until the Nevada is ready for trial.

IMPROVEMENTS IN RECIPROCATING ENGINES TENDING TOWARDS INCREASED ECONOMY AND REDUCTION IN WEIGHT.

The steps taken in pursuit of the above objects are:

1. Increase in steam pressure at engine.
2. Change in design of engine framing.
3. Increase in piston speed.
4. Use of superheat, but to a small degree only.
5. Reduction of clearances in cylinders.
6. Decrease of frictional losses through steam ports.
7. Positive circulation of steam through steam jackets.
8. Reduced back pressure in low-pressure cylinders.
9. Increased ratio between low-pressure and high-pressure cylinders, with consequent increased ratio of expansion of steam.
10. Application of forced lubrication to all journals, crosshead guides, eccentrics and thrust bearings.

While the following improvements, both with reciprocating engines and with turbines, have been made:

11. Improved condensing apparatus resulting in higher vacuum.
12. Rational designs of feed heaters based upon amount of water to be heated and amount of auxiliary exhaust steam available for heating purposes instead of using the old rule of thumbs of allowing a fixed number of horse-powers per square foot of heating surface.
13. Basing steam-pipe design upon actual rate of flow of steam through the pipes as determined by tests in service.

14. Reduction of feed-pipe losses to a minimum.

15. Improved evaporators and other auxiliaries.

In addition to the above the reliability of the machinery plant has been improved by—

16. Adoption of high-speed electric-driven forced-draft fans for battleships.

17. Turbine-driven forced-draft fans for destroyers, and the most important of all—

18. The adoption of oil fuel for both battleships and destroyers.

Considering the above changes in detail, the steam pressures at the main engines since 1895 have been increased gradually from 150 pounds per gage to 265 pounds per gage in the high-pressure valve chest, resulting in decreased size of engine cylinders and in decreased size of steam piping for equal units of power.

The engine framing of the vertical engines first fitted was either of cast steel or built up of steel plates, as shown on Plate 9 (Fig. 251). On several vessels trouble has been experienced with this type of framing, particularly when made of cast steel, and, in addition, the weight was high. The Bureau of Steam Engineering, in order to overcome these faults, designed and adopted the style of framing shown on Plate 10 (Fig. 252) for the Kearsarge and Kentucky, and this style of forged-steel, built-up framing has been adhered to.

Since the adoption of this type of framing, framing troubles are unknown, notwithstanding the fact that the weight of the modern framing per indicated horsepower has been reduced to about 3.3 pounds against $5\frac{1}{2}$ pounds for the old.

Since the design of engines for the Oregon were laid down there has been a gradual increase in piston speeds used, from 900 feet in that class to 1000 feet in the Delaware class. This increase in piston speed has been followed by decrease in weight of the moving parts and has aided in holding down the weight and height of the engine, although the stroke has been increased from 42 inches to 48 inches.

On Plate 8 (Fig. 250) are shown lines giving the maximum and minimum heights of naval engines for different lengths of stroke, also the line of crosshead pin centers and crank-pin centers. These curves are all for engines having the bureau's standard ratio of four to one between the connecting-rod length and crank-pin arm.

In the use of superheated steam the Bureau of Steam Engineering has been rather conservative. At present there are seven vessels in the naval service fitted for superheat, the maximum degree of superheat obtained at the boilers being 85° F., which reduces to about 60° F. at the engines. These figures are for full-power conditions, and an increase in economy of about 6 per cent is estimated to be obtained. At 12 knots, the cruising speed, the saving by the use of superheat hardly exceeds 3 per cent.

The first experiences with the vessels fitted with superheat were far from satisfactory, due to the rapid deterioration of the valves in the steam lines. These valves had cast-steel bodies and cast-steel valve discs with monel metal seats. The erosion and corrosion of the valve discs was very extensive, and in a short period of service it became necessary to replace the cast-steel valve discs with discs of monel metal. This substitution has been satisfactory and no further trouble has been experienced.

The superheat has been used only on battleships fitted with reciprocating engines or impulse turbines for propelling purposes and has not as yet been used on any of the destroyers.

Reduction of clearances, decrease of frictional resistances of steam through the steam ports and reduced back pressures in the low-pressure cylinders have all resulted from one very important change in the design of engine cylinders and valve chests. The original design of these cylinders is shown on Plate 12 (Fig. 254), and the modern improved design on Plate 11 (Fig. 253).

The results obtained by this change from the long, tortuous port to the short, direct port can best be shown by the following table of comparison of cylinder clearances and steam velocities:

	Old Type.	New No. 1.	New No. 2.	New No. 3.
Diameter of cylinders:				
H. P.	32½"	38½"	32"	35"
I. P.	53"	57"	52"	59"
L. P.	2-61"	2-76"	2-72"	2-78"
Stroke of pistons	48"	48"	48"	48"
Per cent clearances:				
H. P.	28	16.17	13.83	13
I. P.	20.8	13.17	12.65	13½
L. P.	18.5	12.485	12.02	12

	Old Type.	New No. 1.	New No. 2.	New No. 3.
Steam velocity, feet per minute:				
H. P.	5,670	6,565	6,723	6,402
I. P.	8,262	6,678	7,883	7,517
L. P.	10,833	10,446	9,947	10,281
Exhaust velocity:				
H. P.	5,480	5,340	5,289	5,078
I. P.	6,670	6,180	5,909	6,087
L. P.	7,450	7,514	7,199	7,298
To condenser	7,390	6,637	6,612	6,500

To further illustrate the gains made by the short, direct ports, attention is called to Plate 4 (Fig. 247), where the back pressures on the low-pressure pistons of several vessels at varying revolutions are shown as curves, the designed revolutions being marked on each curve by a circle. Of these vessels the Delaware is the only one with the modern type of port.

In the last ten years the cylinder ratio of low pressure to high pressure for triple expansion engines has been increased from about 7:1 to 10:1, including clearances. This increase in ratio had been used previously in remodeling the engines of the Cincinnati and Raleigh, with most excellent results. A serious mistake was made, however, in counting too much on the increased expansions obtained by fitting a smaller high-pressure cylinder than that originally installed, the steam pressure having been increased. The new high-pressure cylinder was made 24 inches in diameter and the ratio of low-pressure to high-pressure cylinder changed to about $11\frac{1}{2}$ to 1. While the economy obtained with these engines was most excellent, the high-pressure cylinders were entirely too small, and the engines have never developed the expected power.

By adoption of forced lubrication for the main propelling engines the engine friction has been enormously reduced. All the journals are oil borne so that no metal-to-metal contact occurs. The result has been that the amount of adjustment and overhaul of the main engines has been decreased to a very large extent, and the men who would have been used for this overhaul work can now be used on the auxiliary machinery to good advantage. This decrease in wear of the bearings, and the cushion provided by the oil, has resulted in a

much better maintenance of alignment of the engines, has reduced shocks on the machinery, and has reduced vibration due to these shocks.

In addition, there is considerable saving in oil at ordinary speeds. At high speeds there still exists a heavy loss of oil, due to splashing on the cylinder heads and also to loss by evaporation from the hot surface of the lower heads.

When first fitted, the forced lubrication gave trouble, due to oil being drawn through the low-pressure piston-rod stuffing boxes. In order to remedy this defect stuffing boxes fitted with steam seals have been supplied, and later reports indicate that where the steam seal is properly fitted no trouble of this kind now exists.

That the foregoing changes have produced great economy is amply demonstrated by the results obtained with the machinery installations of the Michigan, South Carolina and Delaware. Plate 5 (Fig. 247) shows curves of indicated horse-power per square feet of grate surface, and curves of square feet of heating surface per indicated horse-power for vessels fitted with the new type of engines and with boilers having superheaters, and also for vessels with the older type of engines and no superheat.

In laying down these curves, the actual generating heating surfaces and superheating surfaces have been retained, and the grates have been increased or decreased in order to have a constant ratio of heating to grate surface of 48.5, this being the ratio for the boilers of the Minnesota. At the same time, the air pressures have been corrected inversely as the ratio of the new grates to the actual grates.

The results obtained are shown as follows:

CURVES OF INDICATED HORSE-POWER PER SQUARE FEET OF GRATE
SURFACE.

- A. With moderate superheat and modern engines.
- B. No superheat and unimproved engines.

CURVES OF SQUARE FEET OF HEATING SURFACE PER INDICATED HORSE-
POWER.

- A. Generating and superheating surface per indicated horse-power with modern engines.

C. Generating surface per indicated horse-power with modern engines and superheat.

B. Generating surface per indicated horse-power with unimproved engines and no superheat.

With the advent of the turbine for marine propulsion, if the full benefit of the new machine was to be realized, a high vacuum in the condensers became imperative. In order to obtain such vacuums, the Parsons Company originated the vacuum augmentor, and this addition to the condensing plant is used extensively in the naval service. In some vessels in the service, in place of the ordinary air pump with augmentor, air pumps of the dual type, as manufactured by Weir, have been fitted, while in other vessels both wet and dry-air pumps have been used.

Of these systems, that with augmentors and also the dual type appear to give the greatest satisfaction in service, and in addition require less weight and space than the wet and dry system. Abroad, a new system, known as the "Kinetic," has been developed, and all reports received concerning it have been very favorable, but no example of this system yet exists in the American naval service. In conjunction with these improved pump systems has occurred an improvement in the tube spacing and the baffling of the condensers in order that a better separation of air from the water of condensation will occur in the condensers.

The improvements in design of feed heaters, steam pipes, and feed pipes naturally followed on the measurements of water consumption of the machinery taken during the acceptance trials of the vessels. These measurements placed in the hands of the Bureau of Steam Engineering data of great value, and that bureau has attempted to use the full value of it in proportioning these important items.

For instance, the feed heaters of the Delaware were, for lack of data, proportioned on the basis of so many indicated horse-power per square foot of heating surface, and the two heaters combined have a total heating surface of 2100 square feet. In her sister ship, the Utah, the same degree of feed heating is obtained with heaters having a total surface of only 512 square feet.

In the search for economy the Bureau of Steam Engineering has adhered strenuously to the use of feed heaters with auxiliary exhaust steam as the heating medium, using any excess of exhaust in the low-pressure turbines or the second receivers of triple expansion reciprocating engines. This utilization of the auxiliary exhaust has not been to the taste of the turbine manufacturers, who prefer to use all of this steam in the turbines, depending for feed heat upon that derived from steam drains discharging into the feed tanks.

That the bureau's contention that greater economy of fuel is obtained by the utilization of the auxiliary exhaust in the feed heaters is correct is shown by the curves given on Plate 3 (Fig. 247). The boilers of the vessels from which these results were obtained were so similar that the efficiencies may be taken equal, and the comparison can then be made on an assumption of 100 per cent efficiency without much error. The fuel oil per knot is obtained from the heat units per knot absorbed by the boilers for the different speeds, and dividing these different heat unit values by an assumed British thermal unit value per pound of fuel. At 29 knots it will be seen that the vessel making full use of her feed heaters required only 84.2 per cent as much fuel as the vessel putting all the auxiliary exhaust into the turbines (closed exhaust) and only 92 per cent as much as the vessel using the feed heaters to a limited extent.

The same comparison is shown on Plate 6 (Fig. 248), where are also shown the shaft horse-power curves for two vessels of similar hull, but one being fitted with two shafts and the other with three. Here at 29 knots the three-shaft vessel, though having a poorer propulsive coefficient than the two-shaft one, only requires 90.4 per cent as much fuel per knot. The shaft horse-power of the two-shaft vessel is only 98.44 per cent of that for the three-shaft one. Had they been equal the fuel for the three-shaft arrangement would have been still lower as compared with the other vessel.

The improvements in evaporators consist mainly in the adoption of double effect connecting and in throwing open the gates to other than the standard bureau design, although these are not the only changes from former practice. Evaporator feed heaters using the vapor from the evaporators as a heating medium have been fitted,

and vapor pipes, better designed for the amount they have to carry, are installed.

Until the adoption of electric-driven blowers for battleships and other large vessels and of turbine-driven blowers for destroyers and small vessels, the successful outcome of any heavy forced-draft run was always endangered by the unreliability of the blowers. Since the adoption of these types of blowers this danger of breakdown has been almost entirely eliminated, and, so far as the destroyers are concerned, the blowers may be classed as one of the most, if not the most, reliable of the auxiliaries fitted.

OIL FUEL FOR DESTROYERS AND BATTLESHIPS.

In deciding to adopt oil fuel for use on battleships and destroyers, the Navy Department took into account the following advantages which would be gained by its adoption:

1. Less fuel required for any given radius of action, consequently less percentage of displacement and less bunker capacity required for the fuel.

2. Increased boiler efficiencies.

3. Decreased fire-room force.

4. Less deterioration of boilers due to maintenance of more even temperatures.

5. Ability to maintain high powers for indefinite periods.

6. Less deterioration of ship's structure due to there being no water or ashes in the bilges.

7. Greater cleanliness.

8. Greater ease in replenishing fuel supply, both in port and at sea.

9. Less floor space required for the development of a given power.

10. Greater ease in control of steam supply.

In opposition to these undoubted advantages the following disadvantages exist:

1. Fuel oil less widely distributed over the earth than coal.

2. Greater unit cost than coal.

3. Greater danger of fire than with coal.

The reply to the first disadvantage is that in time of war a fleet operating far from a base would depend upon fuel ships for re-

plenishing her bunkers, and that oil can be carried in bulk as well as coal, and bases where stores of oil can be kept on hand are as easily established as are bases for coal, and such oil bases would have, in case of danger of capture by an enemy, the additional advantage of being much more readily destroyed, together with their stores of fuel, than are coal bases.

The second disadvantage, that of excess cost over coal, is more than compensated for by the quoted advantages.

The third disadvantage, that of danger from fire, is very thoroughly guarded against by storing the oil in compartments remote from the boiler rooms and situated well below the water-line of the vessel. In addition to these primary precautions additional safeguards are provided which render the danger from fire fully as remote as the danger from magazine explosions.

Upon deciding on the adoption of oil as a fuel for the naval service, the Bureau of Steam Engineering examined carefully all the systems for burning oil that now exist and finally decided upon that of mechanical atomization of the oil as the one most suitable for naval use.

In this system the oil is pumped through heaters to the burners, within which it is given a whirling motion. The small central core of oil, discharging through the tip orifice with this whirling motion, the oil flies off and forms a cone of fine mist. This oil mist mixes thoroughly in the furnace with air which passes into the furnace through a cone register surrounding the burner, the register having adjustable openings and guide vanes so that the amount of air to each burner may be regulated and the direction of flow of this air be slightly oblique to the axis of the cone of oil.

The success of the system depends almost entirely upon the proper handling of the air. Improper air regulation will produce a series of rapid explosions of oil in the furnaces, with consequent destruction of the brick linings of the furnaces. With proper handling the oil burns almost noiselessly, and the amount of smoke produced can be held absolutely under control.

In the first battleships fitted with oil fuel the oil was only fitted as an auxiliary fuel and was intended to be used as an aid in keeping up steam when the coal should be so low as to be remote from the

fire rooms and so require excessive trimming. The results obtained with this mixed system are not to be rated as good, nor were good results expected, as the furnace volumes of coal-burning boilers are too small to permit efficient burning of oil. Furthermore, when burning the oil and coal in combination, it is impossible to so regulate the air supply that each fuel will obtain the proper amount. This results in excessive production of smoke and no increase in steam production over coal alone.

To illustrate the value of oil when used as the sole fuel as compared with coal, Plate 8 (Fig. 250) has been prepared. The two fuels were tried under the same boiler, the grates being removed after the coal trials and the fronts rearranged to accommodate the burners and air cones.

To compare the relative values from this plate, let a ratio of heating surface to grate surface of 40 to 1 be assumed and a rate combustion of 40 pounds of coal per square foot of grate be taken, the coal having a thermal value of 14,500 British thermal units per pound. This rate per square foot of grate gives one pound of coal per square foot of heating surface per hour and a supply to the furnace of 14,500 British thermal units per square foot of heating surface per hour.

From Plate 8 (Fig. 250), where this value of British thermal units per hour per unit of heating surface is supplied to the boiler in the form of coal, the consequent evaporation from and at 212° F. per square foot of heating surface will be 10.8 pounds of water. Should this same amount of British thermal units, 14,500, have been introduced into the furnace in the form of oil fuel, the resultant evaporation would have been 11.72 pounds of water.

One pound of oil, however, may be rated at a thermal value of 19,200 British thermal units, therefore the total evaporation per pound of oil would be $\frac{19200}{14500} \times 11.72 = 15.519$ pounds against 10.8 pounds per pound of coal, a ratio of 1.437 to 1, while, if the thermal values of the fuels be compared, the ratio is 1.324 to 1.

The difference between these ratios, in favor of actual evaporative results, is caused by the elimination of all grate losses and losses due to sooting up of heating surfaces when oil is used.

CONCLUSION.

With such a large field to cover as suggested by the title of this article, the limitations as to length of the article prohibit anything more than a brief discussion of each point considered, but it is hoped that what has been presented will assure the Society of Naval Architects and Marine Engineers that engineering progress in the naval service has not ceased, and that the Bureau of Steam Engineering, greatly assisted by its co-workers in the development of the navy, that is, the other technical bureaus of the Navy Department and the engineers of the various shipbuilding yards engaged in naval work, is to-day at least as progressive and as free from ultra conservatism as it has ever been in its history.

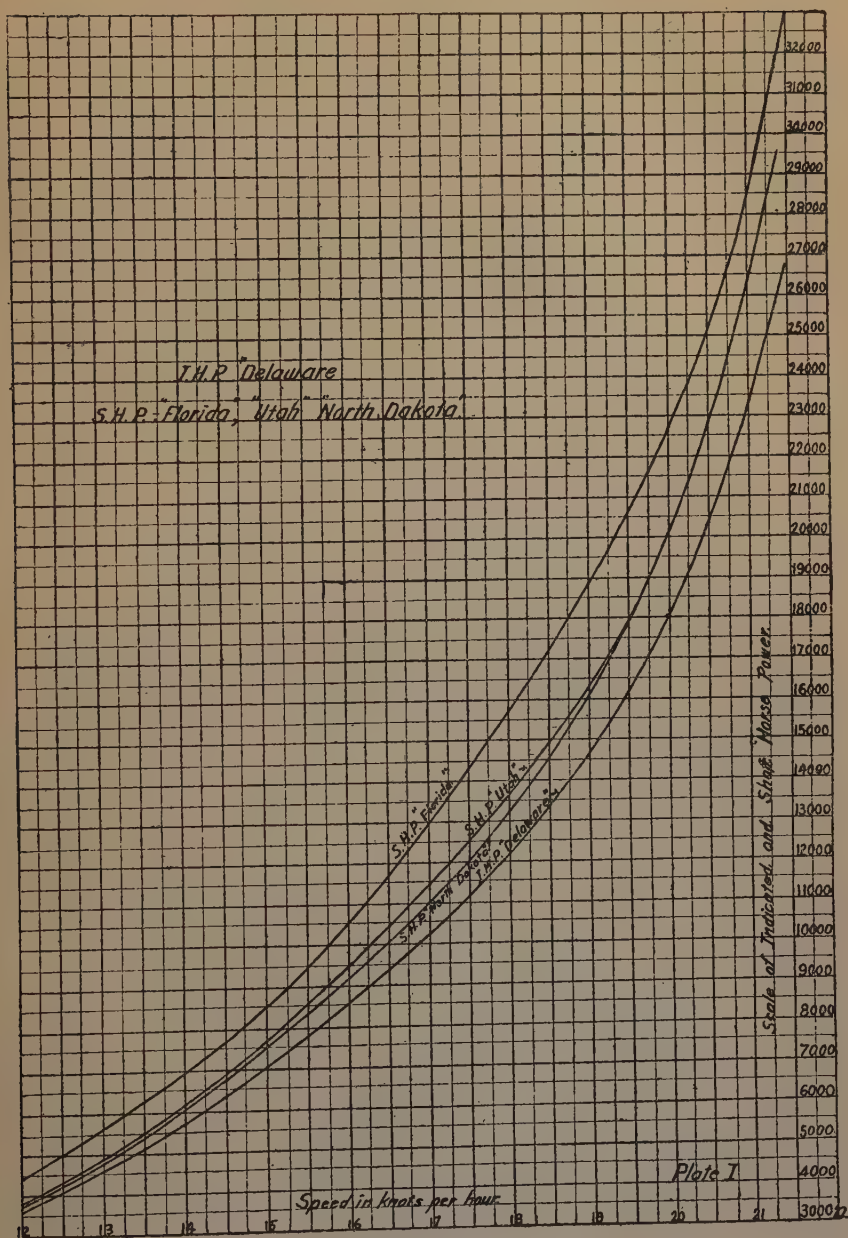


FIG. 245.

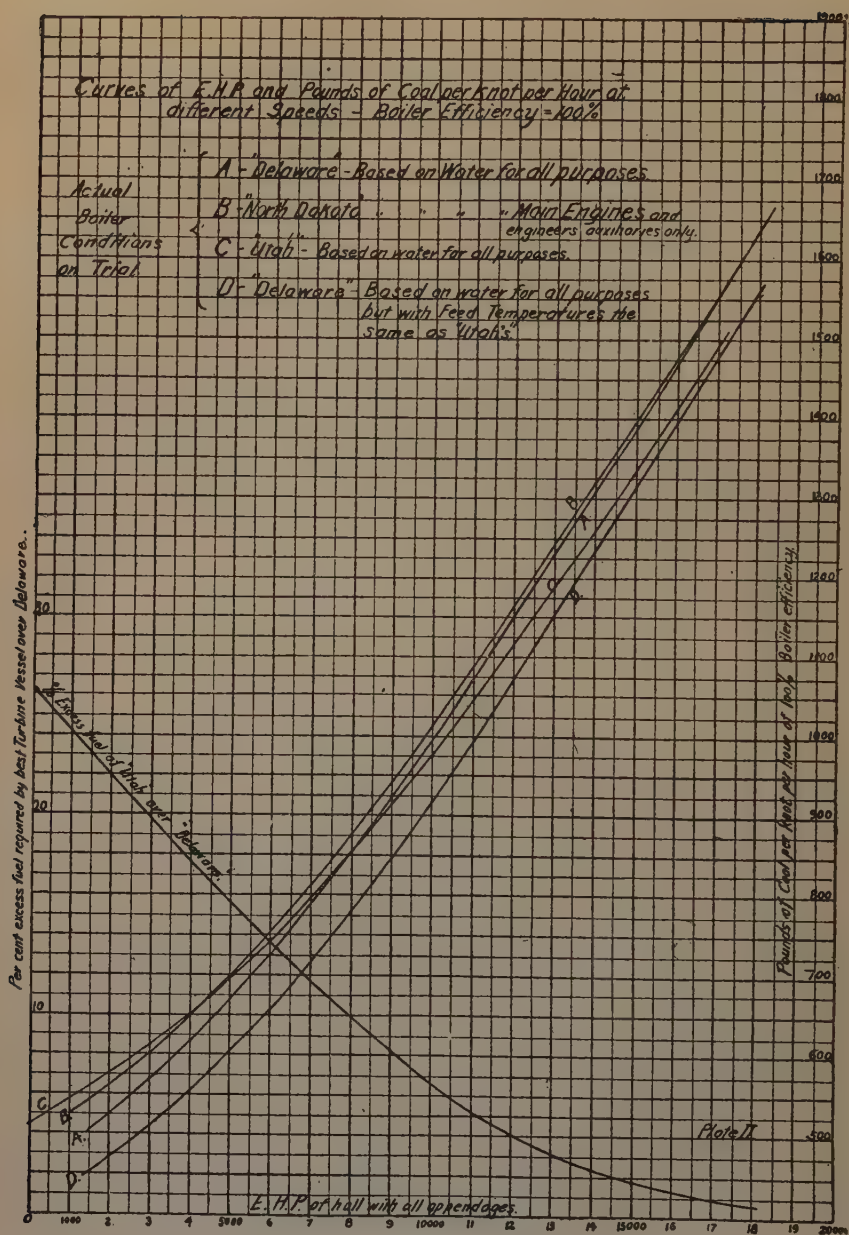


FIG. 246.

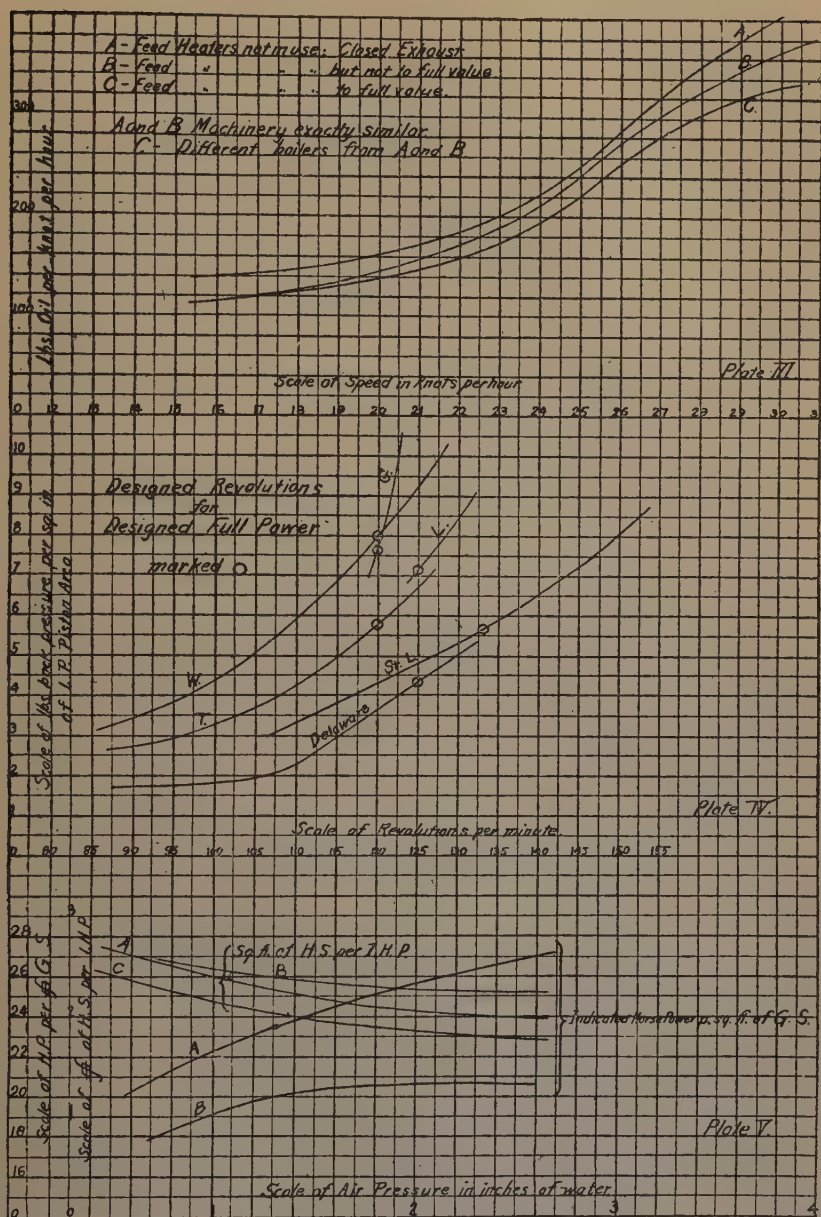


FIG. 247.

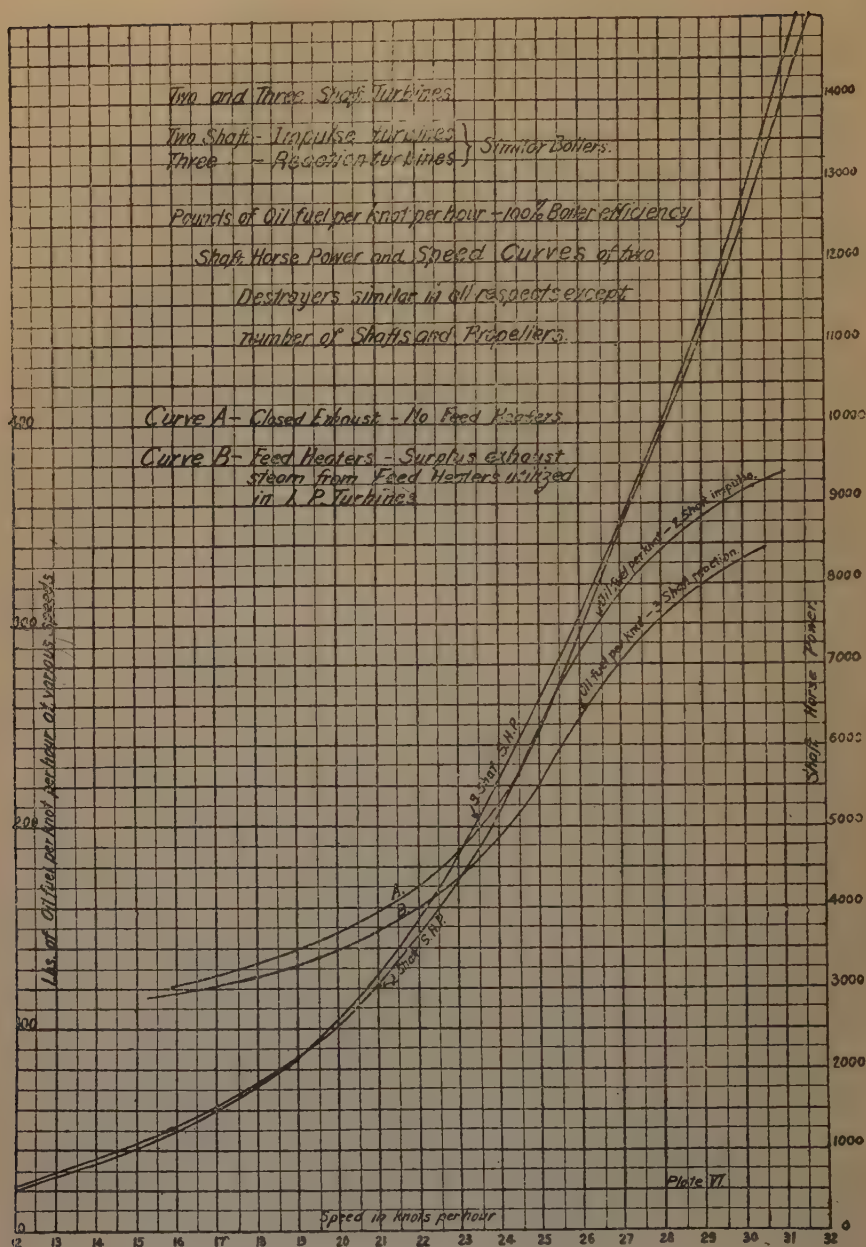


FIG. 248.

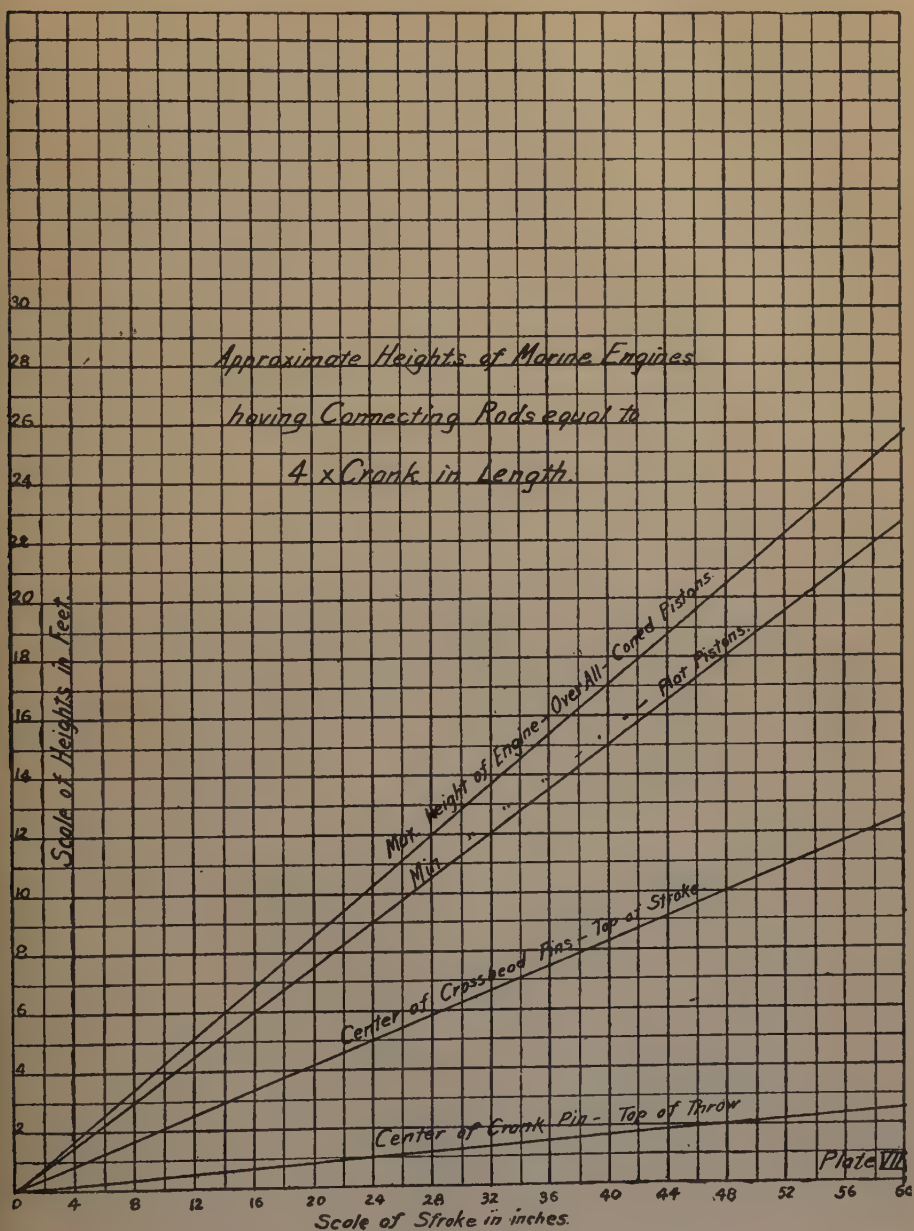


FIG. 249.

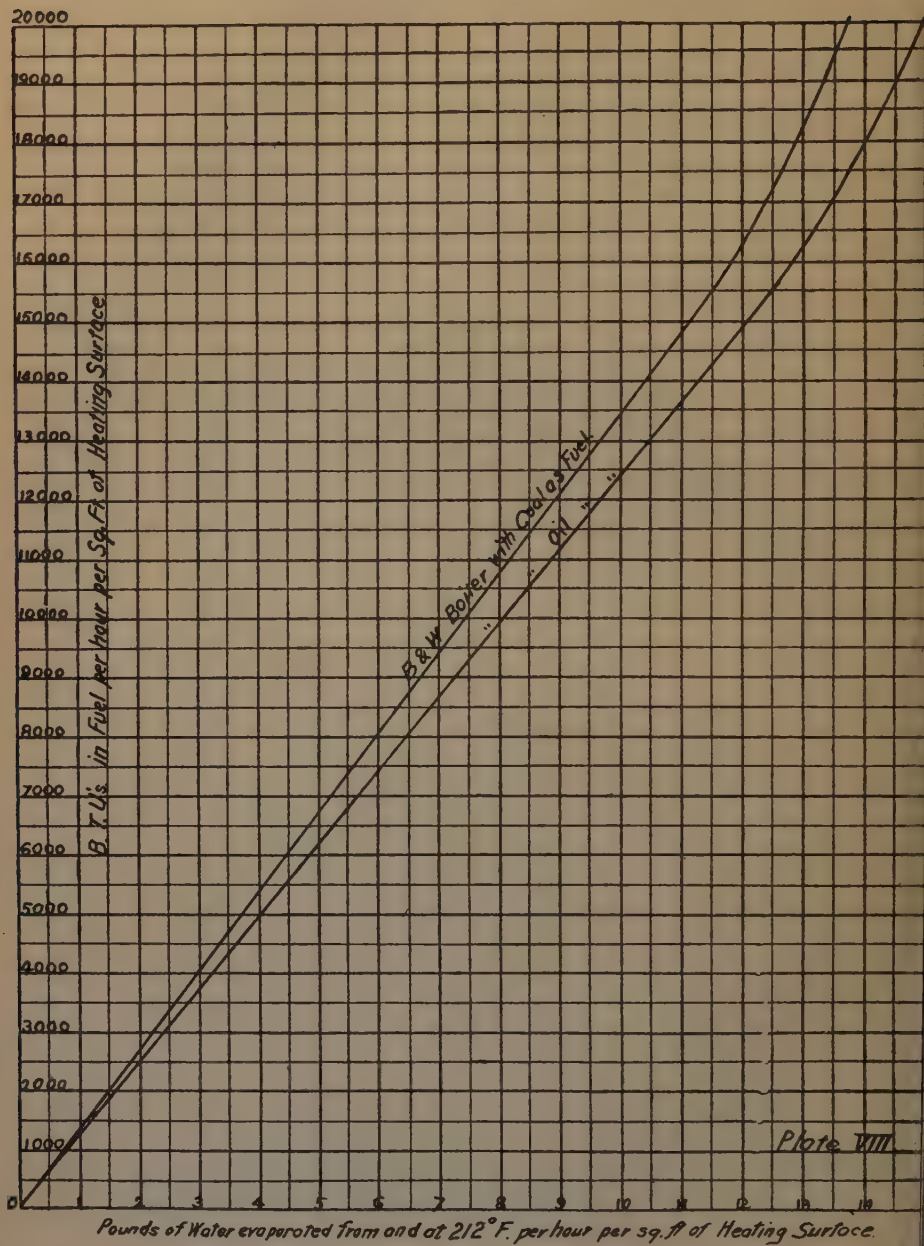


FIG. 250.

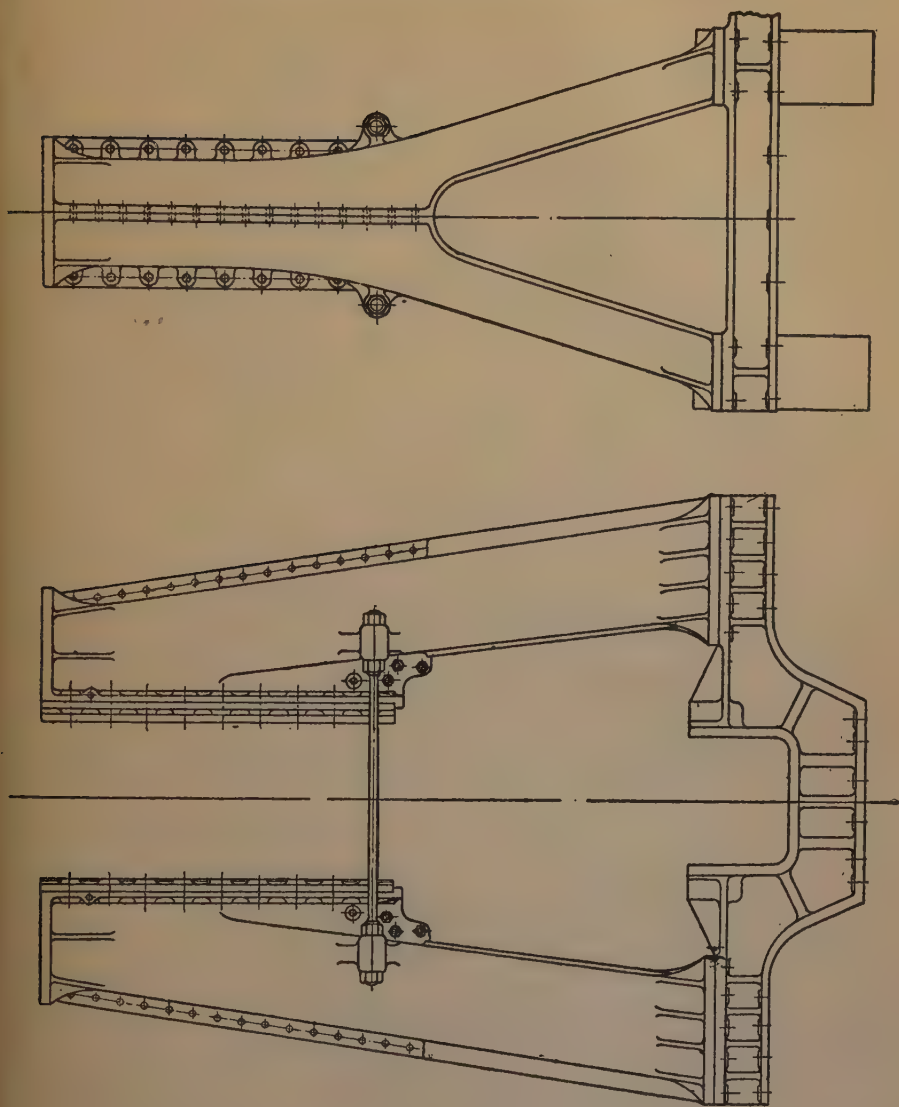


FIG. 251.

U. S. S. Brooklyn, Cast Steel Engine Framing.

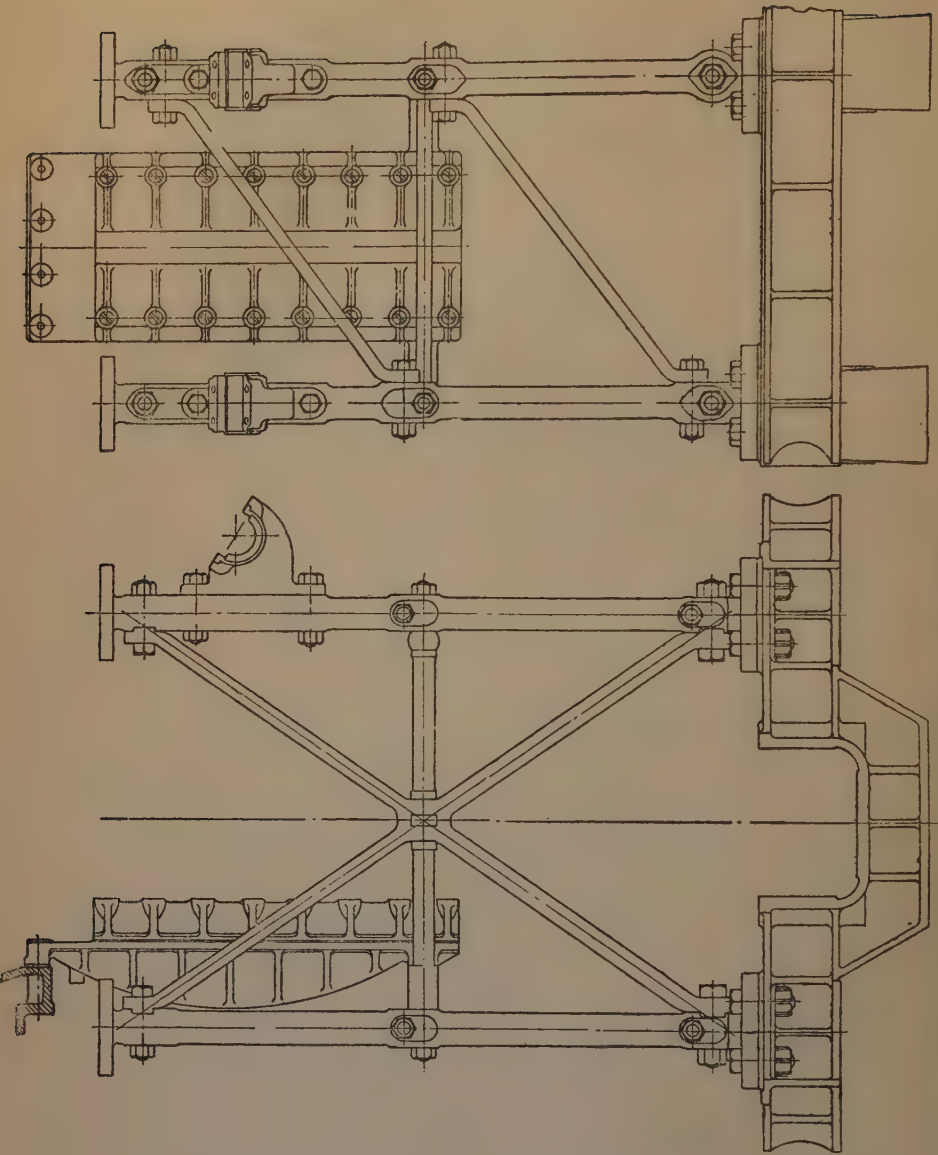


FIG. 252

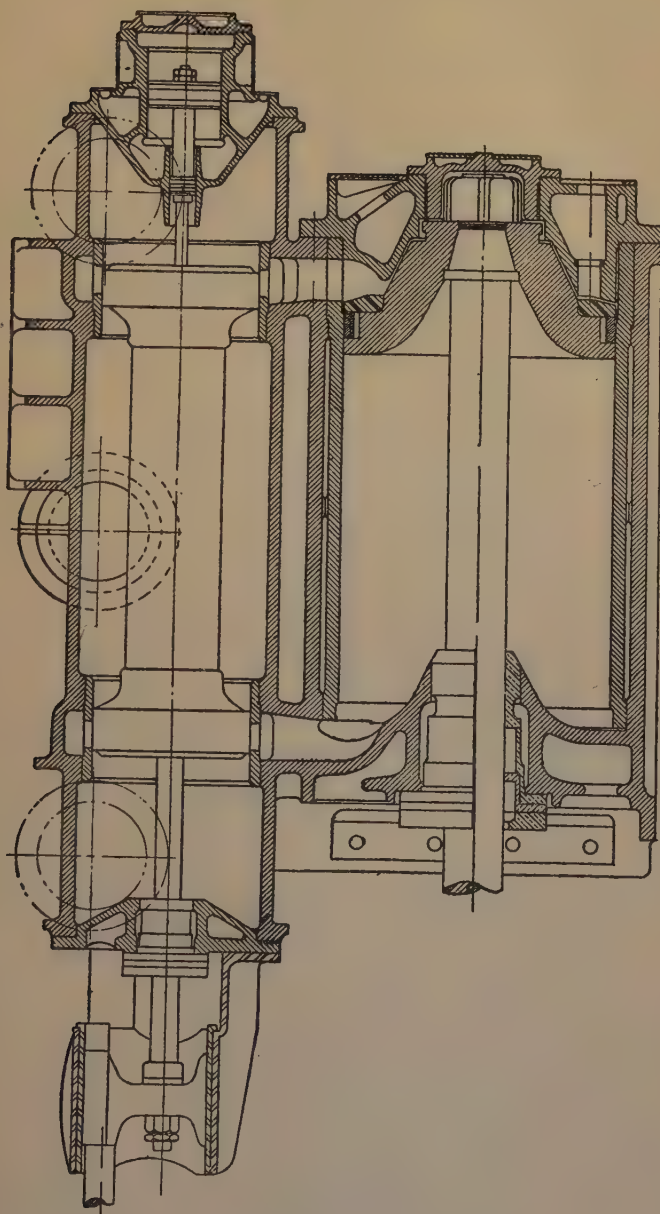


FIG. 253.

U. S. S. Oklahoma, H. P. Cylinder.
Volumetric Cylinder Clearance = 10.75% (mean).

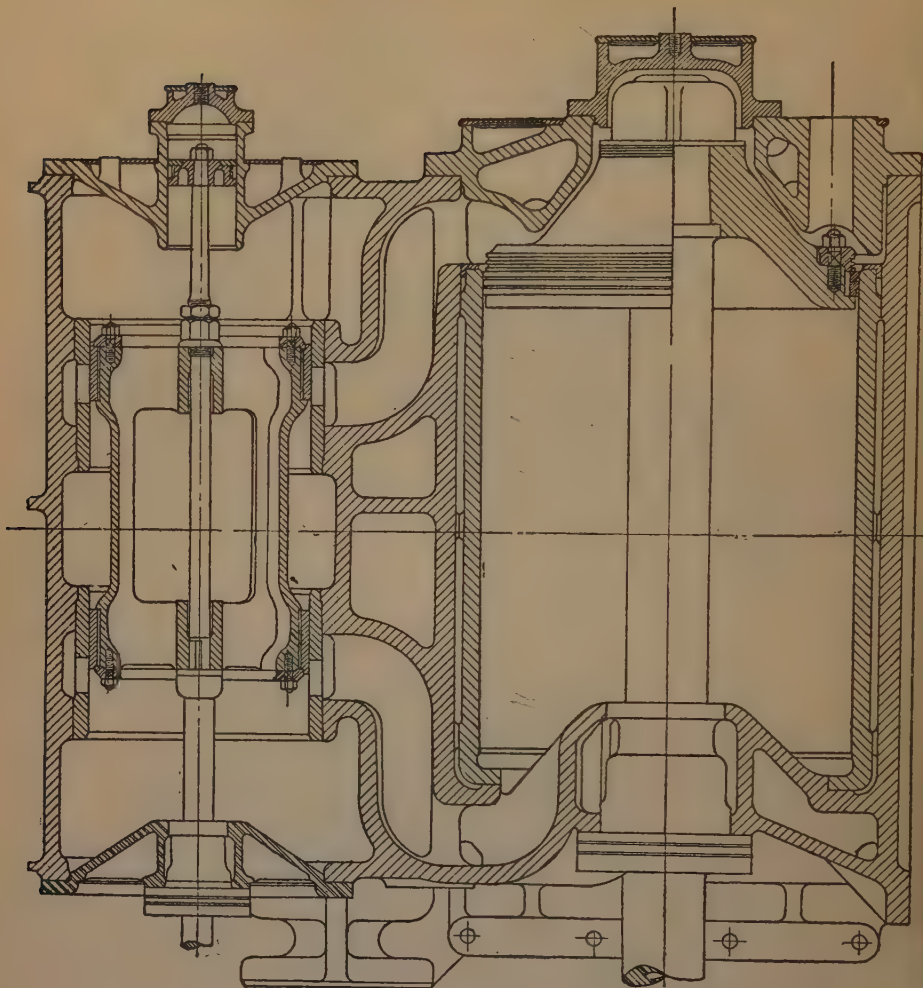


FIG. 254.

U. S. S. Maryland, H. P. Cylinder.
Volumetric Cylinder Clearance = 27% (mean).

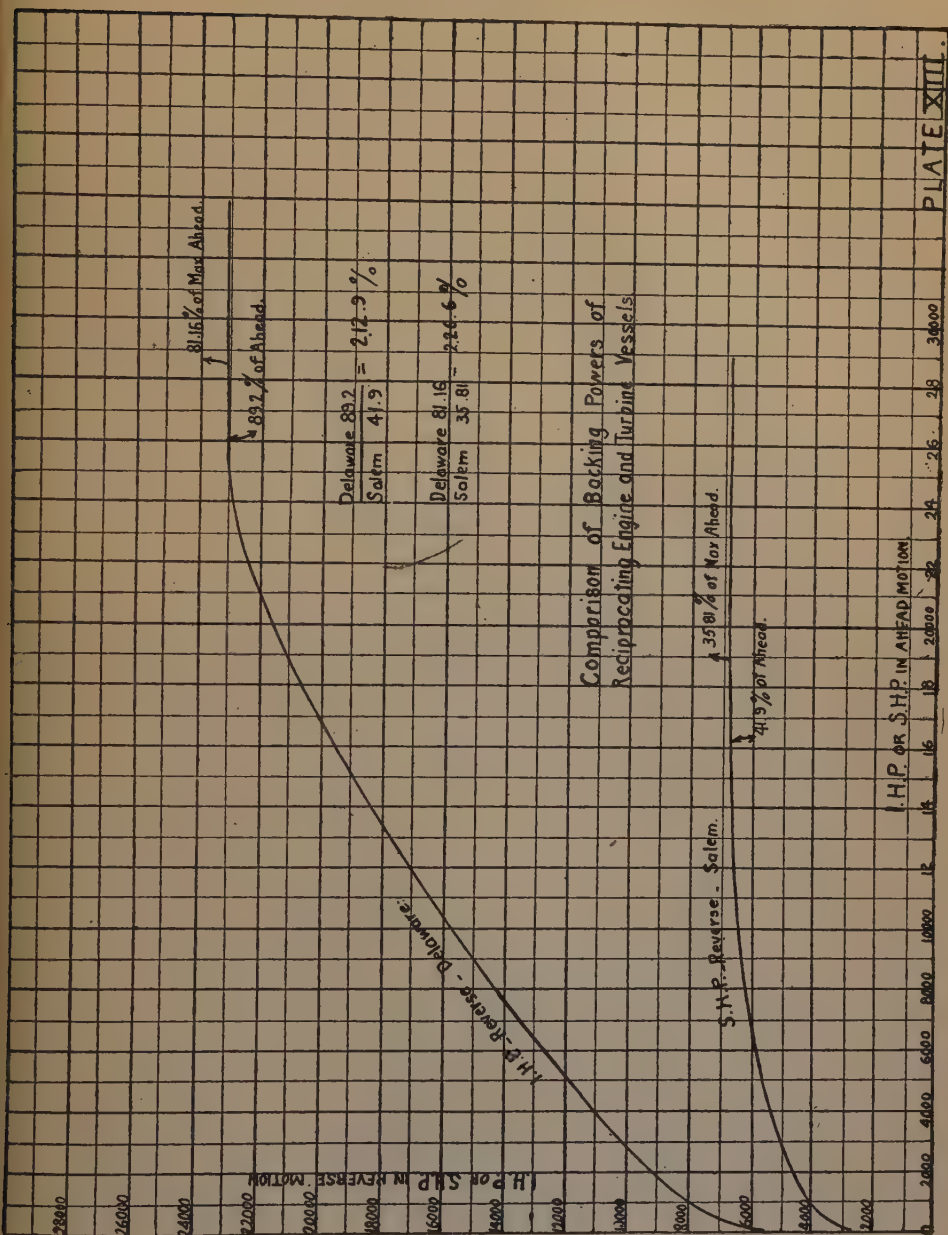


FIG. 255.

APPENDIX.

A

EXTRACT FROM U. S. NAVY REGULATIONS.

SECTION 8 (INSTRUCTIONS).—CARE AND PRESERVATION OF BOILERS AND MACHINERY.

3041. Character of Instructions.—The following instructions for the care and preservation of boilers and machinery relate primarily to the installations in vessels in commission. They shall be considered, however, as applying also to the boilers and machinery of steam launches and motor boats, of vessels out of commission, in ordinary, in reserve, or performing special service, and to shore stations, as far as may be consistent with the character and conditions of the service performed.

3042. Protection and Examination.—(1) Every part of the boilers and machinery shall be maintained in efficient working order, properly protected from undue deterioration, and, as far as possible, ready for immediate use. All internal and external surfaces, working parts, attachments and appurtenances of the boilers, main engines, condensers, auxiliaries, and other apparatus and appliances under the cognizance of the Bureau of Steam Engineering, shall be examined, cleaned, overhauled, adjusted, or tested at regular intervals to insure their good and efficient condition and proper preservation. These operations shall be carried out as opportunities are afforded and as nearly as practicable in accordance with a definite routine so arranged as to fulfill all the requirements of these instructions.

(2) No destroyer's machinery shall be operated at any time with higher pressures than the specified settings of the relief valve on the various parts of the machinery will allow.

3043. Inspection of Cylinders.—(1) As soon as practicable after each run, if the main engines are not likely to be used for a period of three weeks, the cylinders of main engines of the reciprocating type shall be opened and dried out and the walls covered with a thin coating of vaseline or cylinder oil. If, however, it should be found necessary for any other purpose to open any such cylinders after a run, and unless it is expected that the engines will be used again within three days, then such cylinders as it may be necessary to open shall be dried out and coated with vaseline or cylinder oil. In any case all air inlets, such as drains, indicator connections, etc., whereby air might be admitted into the cylinders, shall be kept tightly closed while the engines are lying idle.

(2) The interiors of the cylinders and the piston nuts, rings, springs, followers, and follower bolts of main engines of the reciprocating type shall be examined after each run when they have been worked at or above two-thirds of their designed full power, and also at intervals of not more than thirty steaming days when worked at lower powers.

(3) **Engines to be Jacked Daily.**—When not under steam main engines of the reciprocating type shall be turned at least one complete revolution every day when possible, and the main valves and links moved through their full travel.

3044. Lifting Turbine Casings.—(1) The casings of turbine engines shall be lifted and their interiors examined whenever the commanding officer, upon the recommendation of the engineer officer, deems it necessary or advisable. Whenever this is done advantage shall be taken of such opportunity as may be presented at that time to observe the condition of all nozzles, blading, packing rings, and other internal working parts. When practicable such examinations shall be made when the vessel is where adequate facilities are available for making such necessary repairs as may be found beyond the capacity of the ship's force. The commanding officer shall make a special report to the department (Division of Material) on the condition of the turbines as disclosed by each such examination, which report shall also refer to the date of the last preceding examination and report.

Interiors of Idle Turbines to be Kept Dry.—(2) To reduce to a minimum the corrosion of the interior surfaces of turbine casings and rotors it is of the utmost importance that their interior spaces be kept thoroughly dry while the turbines are not in operation. To this end, after securing the main turbines the main air pumps shall be continued in use maintaining a moderate vacuum until the turbine casings have cooled down to approximately the temperature of the engine room. When it is expected that main turbines will not be required for use again under steam within forty-eight hours, the main air pumps shall be employed daily for a few minutes to produce a moderate vacuum in their interior spaces for drying them out.

3045. Auxiliary Machinery.—All auxiliary steam machinery not in frequent use shall be moved by hand every day, when possible, and by steam at least once each week.

3046. Machinery Securings.—All holding-down bolts, chocks, and ties fitted to the engines, boilers, and other parts of the machinery to prevent them from shifting shall be examined frequently and kept in efficient condition. They shall be carefully tested by sounding or other suitable means to insure their reliability for their particular purposes at least once in three months.

3047. Cold Weather.—Care shall be taken during cold weather to prevent damage being done by freezing to any parts of the boilers, ma-

chinery, or fittings containing water. To this end the temperature of compartments containing such parts shall be kept above 40° F., if possible.

3048. Joints.—All joints, valves, and cocks of the steam pressure and exhaust piping, of the feed suction and delivery systems, and of the condensers and feed tanks, by which water may be lost or the vacuum impaired by leakage, shall be examined frequently and kept tight.

3049. Glands and Packing.—The packing of the glands of all piston rods and valve stems must be adjusted with great care to avoid leakage. Particular attention shall be given to those connected with low-pressure cylinders and other parts working below the atmospheric pressure to prevent not only loss of vacuum but the admission of air to the feed system.

3050. Condensers.—(1) Every effort shall be made to keep the condensers tight, and to prevent the leakage of salt water to the feed system. Whenever the tubes of condensers are found to be leaking, steps shall be taken without delay to locate and stop such leak and prevent the admission of salt water to the boilers. Care shall be taken to prevent water accumulating in the condensers and overflowing into the cylinders of the engines.

Condenser Tube Packing.—(2) Care must be taken that the condenser tube glands fit tight in the threads and are screwed into the tube sheets to a sufficient depth to insure against backing out. For this reason too much tube packing must be avoided.

Condenser Tubes.—(3) The tubes of surface condensers shall be examined at least once in six months and kept clean. If considerable steaming is done, they shall be examined at more frequent intervals. When the tubes and other interior surfaces are found coated with grease, this may be removed by boiling them with a solution of soda. Specimen tubes from different parts of each condenser shall be drawn from time to time and examined to ascertain their condition. An early indication of deterioration is brittleness, and when this state is indicated they must thereafter be watched very carefully, and a special report shall be made describing their condition. Timely requisitions shall be prepared to insure having a sufficient supply of new tubes at hand when partial or complete renewal becomes necessary.

3051. Feed Pumps.—(1) The boiler feed pumps shall not be used for other purposes than those connected with the service of the boilers or feed water, except in cases of emergency; and when not under steam their pistons and valve gear shall be moved every day.

Combined Feed and Fire and Bilge Systems.—(2) In some vessels pumps have been fitted for use either as auxiliary feed pumps or as fire and bilge pumps. In such cases, in order to avoid the possibility either of admitting salt water to the boilers or of wasting feed water, those

pumps regularly used as feed pumps shall have all valves of the salt-water connections wired shut, and those pumps regularly used as fire and bilge pumps shall have all valves of the feed-water connections wired shut, the wiring being removed only in cases of emergency and urgent necessity, or for purposes of examination and test not more frequently than once a quarter.

3052. Evaporators.—The interiors of the evaporators shall be examined, and the tubes or coils cleaned and scaled at as frequent intervals as practicable. All relief valves, gauges, cocks, and other fittings shall be kept in thoroughly efficient condition, and the openings in the shells and pipes connecting them with the interiors shall not be allowed to become choked. Upon the completion of scaling or general overhauling of an evaporator, all parts subject to the pressure of the primary steam shall, if practicable, be tested by steam to insure their tightness. When an evaporator will not be required for use for several days, the shell and coils or tubes shall be drained and kept dry till needed for service.

3053. Zinc Protectors in Condensers, Etc.—Zinc protectors shall be fitted in the water heads of the condensers and in the feed tanks, evaporators, water cylinders of pumps, ice-machine coolers, and all copper suction and discharge pipes designed to convey salt water, in a manner approved by the Bureau of Steam Engineering and shown on the official drawings. They shall be examined from time to time, scaled and refitted to keep them effective, and renewed when they become much deteriorated.

3054. Indicators.—Care shall be taken to preserve the indicators in a clean and efficient state and free from any corrosion or deterioration. They shall not be allowed to remain attached to cylinders when not required for immediate use, and shall be carefully dried and cleaned and lightly coated with cylinder oil before being put away.

3055. Machine Tools, Etc.—All machines, tools, instruments, and other appliances and fittings supplied for the engineer's workshop or for general use shall be kept in good order and thoroughly efficient.

3056. Hydraulic Machinery.—(1) The hydraulic pumps, engines, pipes, and the gear connected therewith, shall be examined frequently, kept in good order and clear of water when not being worked.

(2) The hydraulic engines shall be moved at least twice a month by means of the pumps fitted for the purpose, to prevent the rams becoming set, and to insure their efficiency.

(3) When water necessarily remains in the pipes, the air cocks shall be left open and care shall be taken to prevent freezing.

3057. Compressed Air Systems.—In compressed-air systems, great care must be taken to keep the air suction strainers and air passages of the compressors from becoming choked, the reversing valve motion correctly adjusted, the steam and air piston packing rings, piston-rod stuffing-box packings, and air inlet and discharge valves correctly fitted and tight, and especially all joints, fittings, and connections of the exter-

nal pressure system perfectly tight to prevent waste of the compressed air by leakage. Owing to the small clearances, great care must be taken in adjusting the bearings of the compressors. All parts of the system subject to pressure shall be tested to the full pressure at least once a year. The air cylinders and their passages shall be thoroughly cleaned at frequent intervals by circulating through them a hot solution of lye or soda, followed by a thorough rinsing with hot fresh water, and finally a light coating of oil on the cylinder wearing surfaces.

3058. Idle Boilers.—(1) Boilers when not under steam nor open for cleaning, overhauling, or examination shall be kept quite full of fresh water made sufficiently alkaline to be noncorrosive. They shall be pumped full within twenty-four hours of the completion of steaming and shall be so kept until within twenty-four hours of again raising steam.

Alternative Method.—(2) When it is not practicable to keep boilers full of fresh water, and generally, when it is known that certain boilers will be kept idle for a considerable length of time, they shall be emptied and their interiors shall be dried out as thoroughly as possible. Open trays of as large capacity as practicable and filled to about half their height with quicklime shall be introduced through the manholes into the upper and lower parts of each boiler. They shall then be closed air tight, and special precautions shall be taken to prevent any moisture entering the interiors while they are being thus treated. If necessary, joints of the feed and blow systems shall be broken and adjacent sections of steam piping shall be shut off and their drains left open.

Boilers Open for Work.—(3) Whenever the boilers are open for cleaning and overhauling, their interiors shall not be allowed to remain in a damp condition longer than required to accomplish the necessary cleaning. The cleaning and washing out of the interiors shall be completed as soon as possible after opening, and then the boilers shall be closed at once and filled. If, in order to complete repairs or overhauling of the internal fittings, it is necessary to keep the boilers open for a considerable time after they have been washed out, their interiors shall be thoroughly dried out and kept dry until they can be closed and filled.

Fire Sides of Boilers.—(4) To prevent corrosion while exposed to the atmosphere, especially during periods of wet weather, the fire sides of the tubes and other heating surfaces, fittings, and parts within the furnaces, combustion spaces and uptakes of idle boilers must be kept free from moisture. Light fires in small stoves or pans placed in furnaces or ash pits may be used to dry out empty or idle boilers.

Furnace Doors, Etc.—(5) The furnace and ash-pit doors and the dampers in the uptakes of all idle boilers shall be kept closed. The furnaces of empty boilers shall not be primed. When practicable, the funnels and escape pipes shall be kept covered when all of the boilers connecting to them are idle.

3059. Zincs in Boilers.—(1) Zinc plates shall be suspended in convenient parts of all boilers, especially at points near the entering feed water. The number of plates specified for new boilers is sufficient to provide a total exposed surface, exclusive of edges, in the ratio of three-fourths of a square foot of zinc surface to each one hundred square feet of heating surface. The number and location of zincs shown on the approved drawings of the boilers shall not prevent any change which may appear necessary to better protect the boilers against undue oxidation which may be manifested at any particular part.

(2) Clean plates of rolled zinc only shall be used, and they shall be of the standard dimensions of twelve by six inches by one-half inch thick, or of dimensions that may be cut from such standard plates with the minimum of loss. Worn or defective zinc scrap shall not be recast for such use. When practicable, they shall be fitted in suitable baskets to catch detached pieces should they become disintegrated. Special care shall be taken to insure proper metallic contact between the zinc plates and the stays, lugs, or plates of the boilers to which they are attached. All surfaces and points of contact shall be made bright and shall be firmly bolted together.

(3) The zincs shall be carefully examined at each inspection of the interiors of the boilers. Should they be found considerably oxidized they shall be scaled and refitted, or replaced by new plates if much deteriorated. Plates that have become bent or distorted shall be removed at once as inefficient.

3060. Precautions When Men are to Enter Boilers.—(1) The steam stop valves, feed and blow valves, and any other valves or cocks by which steam or hot water could enter boilers in which men are to work, shall be shut and secured so that they cannot accidentally open or be opened. The engineer officer on duty shall see that these precautions are taken before he allows any men to enter the boilers.

Precautions in Opening Boilers.—(2) Before removing any fittings or parts subject to pressure or taking off the manhole plates of a boiler after it has been under steam, steps shall be taken to insure a complete absence of pressure by opening the air cock and the test and water gauge cocks connecting with the steam space. Whenever boilers are opened they shall be ventilated for a sufficient time to allow all foul air to escape, and no one shall be allowed to enter them until the purity of the air has been ascertained. The possibility of an explosive mixture of hydrogen and air being present when boilers, evaporators, etc., protected by zinc, are opened shall be borne in mind, and steps shall be taken to diffuse the air contained in them before applying an open light.

3061. Deposits in Boilers.—(1) Great care shall be taken to keep the interiors of the boilers, particularly those of the water-tube type, as free as possible from all deposits and from grease or other foreign matter in suspension. Deposits of scale, grease, or other matter adhering to

the heating surfaces, and loose scale or other solid substance lodging in the tubes, besides reducing the evaporative efficiency, are liable to cause overheating and serious injury. Grease and other foreign matter in suspension in the water, even though they may not seriously foul the heating surfaces or restrict the circulation, tend to produce priming and diminish the production of steam and to aid the various processes of corrosion to which the interiors are particularly subject.

Oil and Salt Water in Boilers.—(2) No tallow or oil of vegetable or animal origin shall be allowed to enter the boilers. Salt water shall never be introduced into the boilers except for the purpose of washing out their interiors, and whenever it is used for this purpose the boilers shall afterwards be drained and rinsed out with fresh water and then thoroughly dried, if required to remain open, or closed and filled with fresh water immediately thereafter.

Fresh Water for Boilers.—(3) The boilers shall always be filled with fresh water of as pure quality as can be obtained.

Changing Water in Boilers.—(4) The water shall be retained in the boilers without change as long as practicable, whether they be steaming or idle. Idle boilers shall not be used for trimming ship nor as reservoirs for storing any other water than that intended for steaming purposes. They shall be run down or emptied only when necessary for examination, cleaning, or overhauling, and the water changed only when it has become unfit for further use by reason of being dirty, acid, or salt. Water may, however, be used from fire-tube boilers, for purposes of make-up feed, if necessary when no other fresh water is available when steaming and none can be supplied by the evaporators; but when this is done the water line in the boilers from which such water is taken must not be left at a level among the tubes, and they must be pumped full again as soon as a sufficient supply of fresh water can be obtained.

Feed Water to be Pure.—(5) The feed water shall be maintained as free as possible from impurities, of which the most injurious are sea water, air, and grease. This requires most careful attention to all parts of the entire machinery plant through which the boiler water ever passes, either in the form of steam or water. All devices provided for removing air and grease from the feed water shall be habitually used and shall be kept in the most efficient condition possible. All leaks shall be eliminated from the boilers, piping, glands, condensers, feed tanks, pumps, and from the entire system generally, and the quantity of make-up feed water required reduced to a minimum thereby. Water distilled from sea water is liable to be actively corrosive, hence the importance of reducing the amount of make-up feed to a minimum.

3062. Tests for Alkalinity and Salinity.—(1) The water in all boilers under steam and in the main feed tanks shall be tested daily for alkalinity and salinity. The water in boilers not under steam shall be so tested weekly. The water in the feed tanks shall be examined every

watch, while underway, and every effort shall be made to promptly discover any salt-water leaks that may develop in the condensers. In making the salinity test by the method of determining the quantity of chlorine present it must be borne in mind that, to obtain correct results, the sample of water being tested must first be made neutral or very slightly alkaline, and reagents used for this purpose must not contain chlorine.

Test for Corrosive Properties.—(2) In addition to the tests required by paragraph (1) the water in each boiler shall be tested once each week for corrosive properties in the following manner. A small piece of clean, bright boiler steel shall be suspended in a clear glass bottle filled with water as drawn from the boiler. If, after twelve hours, the piece of steel shows no signs of discoloration or of formation of rust spots, and if the water does not become discolored, the boiler water shall be considered noncorrosive.

3063. Use of Boiler Compound, Etc.—Navy standard boiler compound, sal soda, or lime water shall be used as necessary to maintain boiler water alkaline and noncorrosive. The Navy standard boiler compound shall be used whenever obtainable. If this compound is not used when it is obtainable, the fact shall be reported to the Bureau of Steam Engineering, and the report shall state in detail the reasons therefor. When boilers are being filled with new water, the boiler compound or sal soda shall be dissolved in fresh water and the solution put into each individual boiler in the most direct manner, preferably through a man-hole or some other opening. When boilers are being used under steam, additional compound or soda solution shall be added to the feed water at frequent and regular intervals, adjusting the quantity according to the quality of the feed water. A sufficient quantity shall be used to maintain the alkalinity at the strength necessary to render the boiler water non-corrosive, as shown by the corrosion test.

Care shall be taken not to use so much soda as to cause priming of the boilers. If a tendency to prime be observed, the amount of soda used shall be decreased, or its use wholly discontinued. This condition will generally exist only when there are considerable salt-water leaks, and these shall be promptly discovered and eliminated. When lime is used, the boiler water shall be kept only sufficiently alkaline to show positively that it is not acid. It will usually be found impossible to secure noncorrosive boiler water with lime, on account of the fact that, at boiler temperatures and pressures the water becomes supersaturated with the lime before the noncorrosive state is reached. Preference shall therefore be given to Navy standard boiler compound or sal soda when either is available. As lime is not entirely soluble in water, it shall never be added directly to the feed water or boiler water. Unslaked lime, when used, shall be dissolved as far as possible in cold fresh water, and allowed to stand for some time in a closed vessel, until the insolu-

ble putty has settled. When the liquid solution is left perfectly clear, it may then be drawn off and added to the boilers or feed water as required. Unslaked lime shall be supplied in air-tight tins, and it is absolutely necessary that it shall not be left exposed to the air longer than necessary, as it possesses great affinity for, and readily absorbs, carbonic acid gas.

3064. Inspection of Boilers by Engineer Officer.—(1) The engineer officer shall keep himself fully acquainted with the general condition of the interiors of the boilers and shall satisfy himself, by periodic inspection and by regularly testing the quality of the water, that no material deposits are present, and that the water is maintained in as clean and pure condition as possible and of the prescribed alkaline strength.

Precautions as to Keeping Boilers Clean.—(2) Whenever there may be reason to believe that deposits may be present in the boilers, they shall be cleaned out at the first available opportunity. It must be borne in mind that boilers which contain deposits on their heating surfaces or have grease or other foreign matter in suspension in the water are not in a safe condition for steaming, and it is particularly important that in such circumstances forced draft should not be used except in case of great emergency.

Frequency of Overhauling.—(3) The length of time boilers may be used under steam before requiring to be cleaned out and overhauled depends upon various circumstances with regard to their type and present state of preservation, the particular requirements of the service for which they are used, and the general character and influence of the local conditions under which they are operated. The engineer officer, knowing the particular circumstances under which the boilers in his charge are used, shall be guided by these circumstances in determining the frequency with which it is necessary and practicable to clean and overhaul them. He shall report to the commanding officer whenever he considers it necessary or advisable to open the boilers for examination or cleaning.

(4) Experience with water-tube boilers in vessels in commission shows that, under the usual conditions of operation, the interiors of such boilers should be completely or partially cleaned and overhauled, as found necessary by examination, after having been used under steam for any purpose for a total of about seven hundred hours since the previous cleaning. In special cases, where the general conditions of operation are particularly favorable, this period may sometimes be extended to one thousand hours. When the operating conditions are not good, however, the boilers shall be opened at more frequent intervals, examined to obtain an assurance of the condition of their interiors, and cleaned, if necessary.

3065. Procedure in Overhauling.—Whenever a boiler is laid up for a

complete cleaning and overhauling, the following general operations and procedure shall be carried out:

(a) Clean fire side and overhaul all furnace fittings, brickwork, baffling, and fire parts.

(b) Empty, open, and wash out the interior of the water spaces.

(c) Clean and inspect the water side and overhaul zincs and internal fittings.

(d) Rinse out and close the boiler.

(e) Overhaul all valves, gauges, cocks, and other external fittings.

(f) Examine and repair, as required, all parts of the lagging, casing, and seating.

(g) Apply hydrostatic test for tightness of valves, gaskets, etc.

(h) Test for tightness under steam, including tightness of casing, and adjust safety valves.

3066. Solid Matter in Boilers.—Loose scale or other solid matter in water-tube boilers is extremely dangerous to their safety, and when cleaning the interiors of such boilers, careful examination shall be made before closing them to insure the absence of loose or detachable scale, disintegrated zinc, or other foreign matter from the tubes and from all other parts of the interiors.

3067. Cleaning Boilers After Steaming.—(1) As soon as practicable after the completion of each period of steaming, the ashes and soot shall be removed from the furnaces, ash pans, combustion spaces, tubes, and uptakes, and all accessible parts exposed to fire shall be thoroughly cleaned and overhauled.

(2) Whenever practicable, the tubes shall be cleaned as thoroughly as possible with the air or steam tube cleaners shortly before the fires die out. Otherwise, if the soot be allowed to remain and cool down with the tubes, it is liable to form a scale which will be difficult to remove afterwards.

(3) The steam tube cleaners shall not be used in the boilers except when the fires are lighted.

3068. Examination of Boiler Tubes.—(1) The boiler tubes shall be examined frequently on the fire side. In water-tube boilers of the straight-tube type the lower rows in the furnaces shall be carefully watched and shall be tested frequently by a straightedge or other means to detect excessive deflection. Deflection and swelling of tubes are signs of overheating, and when either state is observed the interior condition of tubes so affected shall be investigated. Internal fouling sufficient to interfere with circulation or proper heat transmission will cause overheating and result in distortion of some kind. Swelling of tubes can always be detected by passing a gauge over them, and very often by hand. Bulged or blistered tubes shall always be renewed. Straight tubes found deflected more than three-quarters of an inch shall be renewed.

Tube Ends Flared.—(2) All tubes of water-tube boilers, except Field tubes, shall be flared at the ends to prevent them from pulling out of the tube plates or headers. The tubes shall extend through the holes in the plates or headers not less than three-sixteenths of an inch, and flaring shall be accomplished to such an extent that the outside diameter of the flare is made materially greater than the hole into which the tube is expanded.

Tubes with Worn Ends.—(3) In examining water-tube boilers, especially those of small bent-tube type, in which, owing to the shape of the tubes, the pressure tends to force the tubes out of the plates, particular attention shall be paid to the attachment of the ends of the tubes. When the projecting end of a tube becomes worn away to such an extent that the wear extends into the rolled part of the tube, or when it becomes materially thinned in the rolled part, it shall be renewed.

Precautions in Renewing Tubes.—(4) When renewing tubes that are secured by expanding, each tube hole shall be carefully examined to see that it is truly cylindrical and to the correct diameter. After the expanding is done, each tube end shall be inspected to see not only that it has been well done on the inside, but also that it is expanded close to the edge of the hole on the outside.

3069. Exterior Parts of Boilers.—(1) The fronts, backs, bottoms, casings, and other exterior parts of the boilers shall be kept free from scale and rust and from contact with ashes; they shall be kept well protected with paint and as clean and dry as possible. The tops of the casings shall be properly protected against corrosion caused by drippings from air cocks, stop valves, or other sources. The spaces at the backs and sides of the boilers shall be kept clear, to allow access to all parts, and nothing wet or combustible shall be stowed over or around them. The use of hose for washing down the exteriors of the boilers or the bulkheads above the floor plates shall not be allowed.

(2) The uptake casings of the boilers shall be kept clean and well painted. The air spaces between the uptakes and their casings shall be examined frequently and any accumulation of dirt therein prevented.

(3) The air ducts and casings of forced draft systems shall be kept free from ashes and rust and well painted.

(4) The funnel guys shall be frequently examined and adjusted, and their turnbuckles kept oiled and in good condition.

(5) The bilges in the firerooms shall be kept dry and well painted.

(6) When it is necessary to keep ashes in the firerooms until a lighter can be obtained they shall not be stowed against any part of the boilers or bulkheads. Boards or heavy canvas shall be used to protect the metal surfaces.

(7) Cotton waste or other materials used for wiping, which have become wet with oil, shall be cleaned or destroyed immediately after using, or shall be put in a covered fireproof receptacle and kept in a safe place until they can be so disposed of.

3070. Engine and Fireroom Gratings.—The gratings over the engine room and fireroom hatches shall not be taken off except in cases of necessity, and shall then be replaced as soon as possible. No material of any kind shall be stowed on or over these gratings, whereby the ventilation may be obstructed or anything may fall through and cause injury to the machinery or the personnel in the compartments below.

3071. Condition of Boilers to be Logged.—All parts of the boilers shall be carefully examined whenever they are exposed for cleaning and overhauling and their conditions as found upon these examinations shall be described in the steam log. Should unusual cases of damage or deterioration be discovered at any time special report shall be made to the Bureau of Steam Engineering, stating in detail the extent of injury sustained, the remedies applied, and, as far as can be determined, the causes.

3072. Reduction of Pressure.—Should the engineer officer at any time consider it necessary or advisable to reduce the load on the safety valves of the boilers he shall report his opinion to the commanding officer, with his reasons. If the commanding officer concurs in such opinion a survey shall be held by a board of not less than two commissioned line officers for the purpose of determining the necessity for the reduction of the load and the amount thereof. Upon approval of the recommendation of the board by the senior officer present the load shall then be reduced in accordance with such recommendation, if found to be necessary or advisable. The amount of the reduction, the reasons for making it, and the load to which the safety valves are then set shall be noted in the steam log and reported to the Bureau of Steam Engineering. The load on the safety valves shall not be increased without the authority of the Bureau of Steam Engineering.

3073. Safety Valves.—The hand gear for lifting the safety valves shall be thoroughly examined at least once each week, whether the boilers are steaming or idle. Whenever steam is raised in any boiler, the safety valves shall be tested by steam and adjusted as necessary to lift at the prescribed pressure. The hand gear shall also be tested at the same time to insure its being in proper working order. These tests of the safety valves and of the hand gear shall be repeated once each week while the boiler is under steam. If the safety valves of any boiler cannot be adjusted to lift properly at the prescribed pressure under steam fires shall be hauled from it and it shall not be used for steaming until the fault has been corrected. A special report shall be made to the Bureau of Steam Engineering in every case where safety valves fail to operate properly. All tests of safety valves shall be recorded in the steam log.

3074. Boiler Fittings.—(1) All valves, attachments, and fittings of the boilers shall be thoroughly overhauled at regular intervals and kept in efficient working condition.

(2) All water-gauge fittings must be kept in a thoroughly efficient

condition. Great care shall be taken to insure that only glasses of proper quality and dimensions are used, that they are correctly fitted in place, and that all cocks and fittings connected with them are examined frequently and kept in proper order.

(3) Boiler steam gauges shall be compared with a standard gauge at least once a quarter, and adjusted to agree with the standard at the working pressure of steam.

3075. Worn Boilers.—Whenever there may be reason to believe that any parts of the main structure of the boilers that are subject to pressure and that involve their safety, except the tubes, are unduly worn or corroded, such parts shall be drilled and their thickness measured. The holes shall not be more than one-half inch in diameter, and afterwards shall be tapped and filled with screw plugs securely riveted over to prevent leakage. The thickness of the parts originally and when thus tested, the probable cause of corrosion or wear indicated, and all other details of the test shall be entered in the steam log and the engineer officer's record book. The exact position of the holes drilled for this purpose shall be marked on the drawings or otherwise shown by sketches for future reference and identification.

3076. Water-Pressure Tests.—(1) The boilers shall be tested by water pressure at such times as the engineer officer may deem necessary or advisable.

(2) Whenever such test is made to prove the safe strength or the tightness of any riveted, expanded, or other permanent structural joints or parts of a boiler, the following method shall be employed. The water shall be heated to a temperature of not less than 150° F.; and, before applying pressure, the boiler shall be completely filled with water and entirely free from air, and necessary precautions shall be taken to insure that there be no leak past the main or auxiliary stop valves into pipes that may contain steam. Where the load on the safety valves is ninety pounds per square inch, or less, the water pressure to be applied shall be double this load; where the load on the safety valves is above ninety pounds and less than one hundred and eighty pounds per square inch, the water pressure shall be ninety pounds greater than this load; where the load on the safety valves is one hundred and eighty pounds per square inch, or above, the water pressure shall be fifty per cent greater than this load. In the case of fire-tube boilers that have been in service longer than two years, the water pressure to be applied shall be limited to twenty-five per cent greater than the load on the safety valves. The pressure shall be increased slowly and be very carefully applied, in order that injury may not be caused by overpressure, particularly if a drill test should have revealed unusual thinness of any parts.

(3) During the application of the water pressure, the boilers shall be carefully examined and proper gauges used, when practicable, to detect any change in form in any of their parts. Should any indications

of probable permanent deformation be observed, the test shall cease, and the weak parts shall be strengthened as necessary. If this be not practicable, a new test pressure twenty pounds below that at which permanent deformation commenced shall be adopted, and the new working pressure shall be that which corresponds to such new test pressure according to paragraph (2). The load on the safety valves shall be reduced to the new working pressure.

3077. Tightness of Valves, Etc., Under Working Pressure.—To prove the tightness of all valves, gaskets and fittings of boilers under the working pressure, the following test shall be made, if practicable, upon the completion of each general overhauling or repair affecting such parts. A water pressure of ten pounds per square inch less than the load on the safety valves shall be applied. After attaining this pressure all connections, including the feed, stop, and check valves, shall be closed and the drop in pressure during a considerable number of hours noted. If the test be made with water of nearly the same temperature as the boiler and the fireroom, the drop in pressure should not exceed twenty pounds in twenty-four hours. If there be no leaks in the boiler or its fittings, there will be no change in the boiler pressure other than that due to change in temperature of the boiler or the water, or both. It shall be borne in mind that leaky feed valves will give false indications and that, until gaskets are softened by heat, there may be slight leaks around the plates, which will readily take up under steam pressure. For the latter reason, whenever sufficient time is available, this test should be made after steam has been raised to adjust the safety valves and the boiler has again cooled down, when this is done in connection with general overhauling. Although hot water searches out leaks with more facility than cold water, the time element included in this test affords opportunity for the water to cool, with consequent contraction in volume and reduction in pressure, giving an appearance of leaks that may not exist. For this reason water used for this test should be as nearly as possible the temperature of the boiler and of the fireroom.

3078. Fuel Oil.—Fuel oil is the residue of crude hydrocarbon oil after the removal, by distillation, of volatile oils and gases. The oil itself is inert, nonexplosive, very difficult to ignite in bulk, and not capable of spontaneous combustion. The vapor from this oil, however, is explosive when mixed with air. This vapor is heavier than air and tends to accumulate in low levels, such as bilges and bottoms of tanks, where it may remain undiscovered until ignited by a naked light or spark. It is always present in a partly filled oil tank, or one that has contained fuel oil and from which the vapor has not been removed by artificial means, and it is expelled through the vents from the fuel-oil tanks while they are being filled. A leak allowed to continue in any part of the oil-burning system may result in an accumulation of this explosive vapor, unless such leak is located in the path of air to the furnace.

Ignition of the vapor may be caused by an open light, electric spark, or spark made by striking metal, heat of the filament of a broken electric lamp, smoking, sparks from funnel or galley, or fires under boilers. An oil fire can be extinguished by sand, steam, or chemical extinguishers, but not by water.

3079. Precautions in Handling Fuel Oil.—The precautions prescribed in section 7, chapter 29, Naval Instructions, regarding the care and handling of fuel oil shall be closely observed.

3080. Precautions in Handling Gasoline.—Instructions for the care and handling of gasoline are set forth in section 9, chapter 29, of these instructions.

3081. Precautions with Internal Combustion Engines.—(1) No smoking, naked lights, or electric apparatus liable to spark shall be permitted in the vicinity of vaporizers, cylinders, or crank chambers of internal combustion engines when any of these parts are open for examination or overhauling.

(2) Pistons, valves, cylinders, and vaporizers of internal combustion engines shall be frequently examined and cleaned, and all moving parts and wearing surfaces shall be kept thoroughly lubricated and in good condition. The pistons and valves shall be kept tight in order that the efficiency of the engine may not be impaired by leakage of air during compression.

3082. Torpedo-boat Engines.—Owing to the lightness of construction and high speeds at which torpedo-boat engines run, increased care is necessary in attending to and adjusting the various working parts. When setting up the brasses of connecting rods, the amount of clearance left at each end of the cylinders shall be accurately measured to see that it coincides with the original clearance. The leads of the valves shall be carefully measured, to determine the necessary readjustment through wearing down of the eccentrics or valve gear. All joints, especially those of the steam pipes, are subject to injury from vibration. The main steam pipes, and any other pipes which appear to suffer in this way, should be tested from time to time to discover any defects arising from this cause.

3083. Machinery of Steam Launches.—Unless it is expected that the machinery of steam launches will be used again in a few days, the wearing surfaces of cylinders and valve chests shall be cleaned and lightly coated with mineral oil, and the engines made ready for use. All drain cocks shall be left open, and the engines, valves, pumps, etc., shall be moved every day. Strainers on the sea-valve openings shall be kept clear, and receiving pipes of circulating and air pumps examined and cleaned annually, or more frequently if necessary. The boilers of steam launches shall be frequently examined, internally and externally, and special attention shall be paid to the furnace. Salt water shall not be

used in these boilers except in cases of great emergency when no fresh water is available, and after its use they shall be scaled and cleaned as soon as possible. The safety valves shall be examined whenever steam is raised after an interval of more than seven days not under steam. The condition of the safety valves, water gauges, check valves, etc., shall be ascertained from time to time while the machinery is working. Great care shall be taken to keep these important fittings in a thoroughly efficient condition.

3084. Docking Ship.—(1) Whenever a vessel is docked, a careful examination shall be made by the yard force of the outboard portions of the shafting, couplings, and propellers. This examination shall be made as soon as practicable after docking, and any defects observed shall be reported at once, so that the necessary action may be taken without delaying the undocking of the vessel. The wear of the lignum-vitæ, both in the stern tubes and the strut bearings, shall be ascertained and recorded. Bronze and monel metal propellers shall be cleaned and polished.

(2) When a vessel is to remain in dock for sufficient time, the shafts shall be disconnected at the engines and the weight required to turn the screws shall be ascertained, as a test to show whether the shaft is free to revolve in the lignum-vitæ bearings and whether the bearings are truly in line. The vessel shall not, however, be detained in dock for the purpose of making this test unless there are grounds for believing that the shafts are bound, and this test need not be made oftener than once in two years. When the test has been made for the first time, the weight required to remove the screw, together with the length of the arm through which it acts, shall be stamped in a conspicuous place on the hub in the following manner: "850 lbs. at 7 ft. radius moved screw—March 12, 1913." If new propellers are fitted, this record shall be stamped on them. On subsequent occasions of making this test, the engineer officer shall cause a comparison of the results of the original and other previous tests to be noted in the log. Should there be any marked difference in these results of the moment required to turn the shafting, the cause shall be fully investigated.

3085. Ships Going Out of Commission.—(1) When a ship is ordered out of commission, the iron or steel bright work of the machinery, except such parts as pass through stuffing boxes, or upon sliding surfaces (as piston rods, valve stems, slide and guide faces, and journals), shall be covered with white lead and tallow.

(2) Metallic packing shall not be removed from piston rods or valve stems, but all vegetable or fibrous packing shall be removed.

(3) All parts passing through stuffing boxes or working upon their surfaces, such as piston rods, valve stems, guide and slide faces, clutch coupling slides, interiors of steam cylinders and valve chests, must be cleaned and covered with a coating of vaseline, the machinery being

moved after first application so as to bring all these parts upon properly covered surfaces.

(4) All bearings shall be well oiled and the oil holes plugged with waste, the engines being turned one complete revolution after oiling.

(5) All water-containing parts of the machinery inside of outboard valves shall be thoroughly drained. Particular attention shall be paid to draining of pump cylinders; condensers; feed, blow, and suction pipes; fire main, and all steam and exhaust piping where it is possible for water to gather. In draining these pipes, flange joints shall be broken at the lowest parts of each system and wherever a pocket is formed which is not drained by a proper drain pipe. Where possible, outboard valve casings below valve seats shall be covered with nonconducting material, such as sawdust or manure, temporarily boxed in, and the interiors of the casings below the seats filled with heavy oil, injected if practicable through the steam connection, to displace the water.

(6) The gauges and oil cups shall not be removed.

(7) The sea valves shall be closed and properly secured.

(8) The storerooms shall be cleaned.

SECTION 9 (INSTRUCTIONS)—OPERATION AND MANAGEMENT OF BOILERS AND MACHINERY.

3101. Character of Instructions.—The following instructions relate especially to such matters as are essential in the operation of the boilers and machinery to prevent injury, to insure their safety, and to secure the most efficient and economical performance. They shall be considered to apply to the same extent as the instructions contained in Section 8 of this chapter.

3102. Cylinders.—(1) The cylinders, receivers, and steam jackets of all engines shall be gradually and thoroughly heated before steam of full pressure is admitted to them. Whenever practicable the connections between the boilers and main engines shall be opened as soon as fires are lighted to allow the hot air and steam as it rises to circulate through the engines and warm them gradually.

(2) Water must not be allowed to accumulate in the receivers or jackets of engines, but care shall be taken that steam is not blown through the traps and wasted while the engines are working.

3103. Water Rams.—The greatest care shall be exercised to guard against water rams by carefully draining all pipes or other spaces where any water might accumulate before steam is admitted to them. In opening the valves connecting the boilers to the steam pipes or in connecting different sections of steam piping, the by-pass valves, if fitted, shall be opened first, and the pipes warmed and the pressure equalized before the large valves are opened. The drains shall be kept open until it is certain that the pipes are entirely free of water.

3104. Air and Circulating Pumps.—Independent air and circulating pumps shall be started at least fifteen minutes before attempting to move the engines, or to warm them by admitting steam to the cylinders through the by-pass valves or otherwise.

3105. Telegraphs, Etc.—The engine telegraphs and other instruments fitted for transmitting signals to and from the engine rooms shall be carefully examined and tried before getting underway.

3106. Rate of Expansion.—(1) Modern engines are designed to carry out the principle of expansion to its utmost practical limits, and, in order to attain the greatest economy, especially when working at reduced speeds, every means fitted shall be utilized to secure the highest possible ratio of expansion of which the engines are capable. To this end, the engineer officer shall ascertain the most efficient rates of expansion at all powers and the corresponding points of cut-off and other particulars of adjustment. These particulars, together with related data, shall be carefully recorded, so that they may be readily utilized when changes in speed or power are required.

(2) In the design of modern reciprocating engines, the cylinder ratios and the details of the valve motions are calculated to fix the ratios of expansion throughout the whole engine required to obtain the best practical economy when working at full power. To provide for varying to some extent the ratio of expansion of the engines as a whole, and of the several cylinders, particularly when working at reduced powers, in order to secure the most advantageous expansion and distribution of work in the several stages of expansion, the links are suspended from adjustable blocks by means of which the point of cut-off and consequently the ratio of expansion in the several cylinders may be varied within small limits. However, in those engines where the lowest stage of expansion is carried out in two cylinders of equal size instead of in a single cylinder, the design does not contemplate the total work of the whole engine being divided equally between all of the cylinders, but rather an approximate equality between the several stages of expansion. It should be borne in mind that the total ratio of expansion throughout the engine may be changed by altering the cut-off for the high pressure cylinder only, and that any alteration of the cut-offs for the other cylinders merely affects the distribution of work between the several stages of expansion, and this only by relatively small amounts.

(3) It is not anticipated that, in effecting the greatest change of ratio of expansion possible within the scope of the adjustable cut-off block for the high pressure cylinder, any inequality of distribution of work produced would interfere with the smooth working of the engine, if the bearings are in good condition and properly fitted. The practice, therefore, of distributing the work equally in each cylinder or stage of expansion shall be considered as of only secondary importance.

3107. Internal Lubrication of Cylinders.—(1) No tallow or oil of vege-

table or animal origin shall be used for the interior lubrication of the steam cylinders and valve chests, and as little as possible of any kind of oil shall be used for this purpose. (This prohibition shall apply to every cylinder and valve chest for whatever purpose used.) Under ordinary conditions of working with saturated steam the water of liquefaction derived from the steam furnishes ample lubricant for the internal working parts, but, if this does not prove sufficient, pure mineral oil only shall be employed.

Lubrication of Piston Rods and Valve Stems.—(2) Care shall be taken that the oil used for lubricating the piston rods and valve stems is not drawn into cylinders or valve chests. When main engines are fitted with forced lubrication systems care shall be taken to prevent the oil from being splashed on piston rods or valve stems, particularly from crossheads or guides.

Grease Extractors.—(3) When filters or grease extractors are fitted they shall be used, except when under repairs or being cleaned. The material used in filters and grease extractors for extracting the oil from the feed water shall be cleaned or renewed as often as necessary to keep it effective and to prevent it becoming saturated or clogged with grease and inefficient. Every available means shall be employed to prevent the passage of oil to the boilers.

3108. Water on Bearings.—Water shall not be used unnecessarily on the bearings, but when it is used care shall be taken that it is discontinued a sufficient length of time before the engines are stopped to allow the lubricating oil to find its way to all parts of the bearing surfaces. Bearings shall not be allowed to get too warm before resorting to the use of water, if such use should become necessary, but it shall be applied at first with the greatest caution in order to avoid cracking or warping any parts as a result of too sudden cooling. Bearings shall be examined at the earliest opportunity after water has been used on them.

3109. Alternation in Use of Boilers.—(1) As the number of boilers required for ordinary cruising and for port service is usually less than the whole number fitted, the employment of the several boilers shall be regulated so that eventually the work will be distributed equally among the whole number, unless some peculiarity of fitting or other similar reason prevents such equal distribution being accomplished.

Forced Draft with Fire-Tube Boilers.—(2) When it is necessary to increase the speed of a vessel having cylindrical fire-tube boilers it shall be done, except in case of emergency, by increasing the number of boilers in use, under natural draft, until the entire number on board are in use, if requisite. Forced draft shall not be used on boilers of this type except in emergencies and during the power trials specified in sections 1 to 7 of this chapter.

Training Firemen.—(3) In order that the best results may be obtained when the development of the highest power is a matter of great impor-

tance, frequent opportunity shall be given for training the firemen to work the boilers at their full capacity, under both natural and forced draft conditions. With this object in view, and to insure that the boilers in use are being worked at approximately their full capacity, when more careful firing will be necessary than is required under easier or more economical conditions of steaming, no more boilers shall be employed upon such occasion than are required for the speed ordered.

3110. Handling of Fires.—(1) Particular attention shall be given to the training of the firemen, especially as regards the management of the fires; and all engineer officers and fireroom petty officers shall take advantage of every opportunity to instruct the firemen how to burn the fuel in the most economical manner. Every effort shall be made to keep the steam pressure and the water level in the boilers constant, to work the fires in the most efficient and systematic manner, and to use to the best advantage all appliances for timing the operations of firing, regulating the supply of air, and for economizing in any way in the expenditure of fuel. The engineer officer shall ascertain the most economical rate of consumption of fuel, together with the number of boilers it may be necessary to employ for any required speed and condition of steaming.

(2) When burning coal, careful attention shall be given to the management of the fires, to secure the utmost economy and efficiency of combustion. The fires shall be maintained at a uniform thickness in all parts of the furnace. Green coal shall be added to the fire at regular and frequent intervals, and shall be scattered over the entire surface. The furnace doors shall be kept open only the shortest possible time. Holes in the fire or the accumulation of clinkers in any part of the furnace shall be prevented. All lump coal must be broken up before being fired. The fires shall be cleaned at regular and frequent intervals, as often as may be necessary to keep them in good condition. Care shall be taken to remove all clinkers adhering to the grate bars. The necessary cleaning of fires shall be done as quickly as possible, in order to reduce to a minimum the amount of cold air admitted through the uncovered grate and the furnace door. The uptake dampers shall be closed while cleaning fires. The uptake dampers, rather than the ash-pit doors, shall be closed when necessary to temporarily check the rate of combustion, the closing of the ash-pit doors being liable to cause the burning or buckling of the bearer bars and grate bars. The use of water in the ash pans is unnecessary under ordinary conditions, and shall not be resorted to except when necessary to prevent clogging of the grates by excessive clinkers.

3111. Temperature of Boilers.—Sudden and very rapid changes of temperature in the boilers shall be avoided. The tubes shall not be unnecessarily exposed to cool air by opening the connection or casing doors; these doors shall not be used as dampers to check the production of steam.

When circumstances permit, at least six hours shall be occupied in raising steam from cold water in other than water-tube boilers. In water-tube boilers, except when the brickwork is quite new, steam may be raised much more rapidly; but sufficient time should be allowed for thoroughly warming up the engines before attempting to move them under steam, and steam may ordinarily be raised in these boilers within the time necessary for this purpose.

3112. Starting Fires.—(1) Before starting fires in any boiler, all drain cocks, bottom and surface blow valves, handhole and manhole plates, and other valves and fittings not intended to remain open, shall be examined to see that they are tightly closed. The safety valves, boiler stop valves, feed check and stop valves, and water column valves shall be examined, and the water gauge and test cocks shall be tested, to see that they are all in proper working order. It shall be definitely ascertained that the valves and pipes leading to the pressure gauges are wide open. The water shall be brought to a height that is slightly below the normal steaming level. The air cock shall remain open while the water is being run down and while steam is being raised, and shall be closed after steam has formed.

(2) While steam is being raised in a boiler, close attention shall be given to all the boiler fittings and feed arrangements, to insure that they are in all respects in proper working order. Special care shall be taken, in setting up the nuts of handhole and manhole fittings, that no greater leverage is applied than that afforded by the proper spanner provided for that purpose. Ash-pit doors of automatic or balanced type shall always be left mounted while boilers are under steam. The boilers shall be connected to the steam line only when there is a difference of pressure not exceeding ten pounds, and the boiler stop valve shall at first be only slightly opened, to allow the pressures in the steam line and in the boiler to equalize gradually. After the pressures have equalized, the stop valve shall be further opened gradually to such extent as required.

(3) Whenever steam is raised in a boiler, in order to insure that the safety valves are in good working order and to ascertain the exact pressure at which they will lift, the steam pressure shall be allowed to rise until these valves should lift if properly adjusted. This may be done after the boiler has been connected, if more convenient, but the boiler shall not be continued in use unless the safety valves have been correctly adjusted.

3113. Use of Blowers.—(1) The fireroom blowers may be run at any time at moderate speed for purposes of ventilation or to assist the draft.

“Natural Draft” and “Forced Draft.”—(2) When burning coal alone, the term “natural draft” implies that, if necessary or desirable, the blowers may be run with the firerooms either open or closed to assist the draft and to supply sufficient air to the fires to support the com-

bustion of not more than twenty pounds of coal per square foot of grate surface per hour. The term "forced draft" implies that the blowers are used to produce the greater draft pressure necessary to force through the fires the quantity of air required to support the combustion of more than twenty pounds of coal per square foot of grate surface per hour.

3114. Banked Fires.—Banked fires shall not be kept in water-tube boilers; nor shall heavy banked fires be kept in any other type of boilers except in emergencies, and in such cases ash-pit doors shall not be closed. When the main engines are not to be used for twenty-four hours or more, fires shall be allowed to die out in all water-tube boilers, except those needed for auxiliary purposes, and when the main engines are to be used again within twenty-four hours light fires shall be kept spread in such boilers.

3115. Hauling Fires.—Fires shall not be hauled, except to prevent damage to a boiler in case of emergency. When steam is no longer required, the fires shall be allowed to die out in the furnaces, with the dampers, furnaces, and ash pits closed.

3116. Automatic Ash-pit Doors.—When boilers are under steam the automatic ash-pit doors must not be opened so wide as to prevent proper automatic closing in case of any injury causing undue pressure within the furnaces, except during such times as may be necessary for cleaning fires or hauling ashes, upon the completion of which they shall be replaced in their automatic working position. Fire tools shall not be left in the ash pans of steaming boilers.

3117. Feeding Boilers.—(1) In feeding the boilers, the feed valves of the several boilers in use shall be so adjusted as to distribute the requisite supply about equally to each boiler. Strict attention shall be given at all times to maintaining the height of water in the boilers at as nearly as possible a constant level. As long as a boiler is furnishing steam the feed supply shall never be entirely shut off, even for a short period. Should any difficulty be experienced in feeding a boiler, the combustion shall be checked at once, by closing the dampers and ash-pan doors, if necessary, and steps taken to find the cause.

(2) The temperature of the feed water entering the boilers shall be maintained as high as possible. Where feed-water heaters are not fitted, the temperature shall be as high as is consistent with the maintenance of a fair vacuum.

3118. Low Water.—(1) Low water in a steaming boiler is one of the most serious and most frequent emergencies that arise in a fireroom, and is liable to cause distortion of the tubes or other injury to heating surfaces, serious steam and water leaks, or the explosion of the boiler. It is generally the result either of inattention on the part of the water tender, or of his attention being diverted to other duties. Failure of the feed pumps, leaks developing in the feed discharge line, hot or low

water in the feed tank, a defective check valve, or water gauges giving false indications due to defects of the cocks fitted to them are other causes which may lead to low water, if not discovered and corrected with sufficient promptness.

(2) Whenever the water in any water-tube boiler falls below the lowest try cock and out of sight in the gauge glasses the fires shall be hauled, or, if burning fuel oil, the supply of oil to the burners shall be shut off, the safety valves shall be opened cautiously to relieve the pressure gradually and the feed check and boiler steam stop valves shall be closed. The dampers and the furnace and ash-pit doors shall also be closed. All air leaks to the furnace around these doors shall be stopped. Under these conditions the boiler will gradually cool, and any parts that may have become overheated will be subjected to an annealing process. No attempt shall be made to restore the normal water level by increasing the supply of feed water. Fire extinguishers, if fitted, or otherwise a fire hose or wet ashes, shall be used to quench or deaden coal fires before hauling them.

3119. Serious Steam Leaks.—Whenever a large steam leak occurs in a boiler, from the sudden leaking of a tube or other cause, the following action shall be taken, as far as the particular circumstances admit, to prevent serious injury to personnel and to reduce to a minimum the extent of damage to the boiler. The safety valves of the injured boiler shall be opened to relieve the pressure as quickly as possible. The fire extinguisher shall be opened cautiously, if fitted; or if burning fuel oil, the supply of oil to the burners shall be shut off. The furnace and ash-pit doors shall be kept closed until after the steam pressure has been reduced to less than fifty pounds, when, if the fires have not been put out by the leak or by the fire extinguisher, they shall be deadened by a fire hose or wet ashes and then hauled. After the fires are out the doors shall be closed and all air leaks to the furnace stopped and the boiler allowed to cool slowly. The stop valve on the injured boiler shall be closed. If the blowers are running they shall be kept going and their speed increased, if necessary, to drive the escaping steam up the smoke pipe and keep it out of the fireroom; hence, the dampers shall be kept open. Except in the case of low water when consequent overheating is involved, the feed supply shall be continued until the fires are out, to prevent the heating surfaces becoming uncovered and burned; in such case, the auxiliary feed pump shall be started and after the auxiliary feed check has been opened, the main feed supply shall be shut off, if other boilers are being fed from the latter. Special care shall be taken to maintain the water at the proper height in all other boilers in use and to provide additional water from the reserve tanks, if necessary, to prevent a shortage in the main feed tanks.

3120. Accidents in General.—In all cases of accident to boilers or machinery, every endeavor shall be made to localize the injury. The

compartment involved shall be isolated to prevent escaping steam getting into other compartments and interfering with proper attendance on other boilers or machinery in use. All men on duty shall remain at their proper stations, give strict attention to the machinery in operation, and avoid that inattention sometimes due to excitement which may lead to further damage. When considerable leaks of steam occur in a fireroom the upper part of the compartment generally becomes filled with steam, and men shall not be allowed to go up the fireroom ladders at such times on account of the great danger of their being seriously injured or overcome by inhaling the steam. The best avenue of escape, if it becomes necessary to abandon the compartment, is to a coal bunker or to another compartment on a low level.

3121. Economy of Auxiliaries.—With a view to effecting the utmost economy in fuel consumption for auxiliary purposes the following requirements shall be observed:

(a) Steam shall be kept fully shut off from all engines, steam and exhaust pipes, etc., when not required for use. If any additional steam power beyond the usual requirements be needed for temporary use due notice shall be given in advance, and also as soon as the necessity ceases, in order to avoid keeping the steam at an unnecessarily high pressure, an engine working when not needed, or an uneconomical number of boilers in use.

(b) The use of capstan engines, winches, ash hoists, deck pumps, sanitary pumps, and galley, pantry, or radiator heating systems, etc., shall be limited to the minimum necessity of their legitimate functions.

(c) Electric lights, fans, or other electric apparatus requiring the expenditure of power shall not be left turned on when not required for immediate use.

(d) The issue of fresh water shall be under strict supervision to prevent waste.

3122. Oil-Burning Installations.—(1) The principal points requiring special attention in working fuel oil installations are:

(a) The oil pressure: This in great measure governs the rate of burning the oil and shall be maintained as nearly constant as possible. As far as practicable, the pulsation from the pumps shall be removed from the oil before it reaches the burners. This is accomplished by the use of air chambers fitted on the discharge side of the pumps and elsewhere in the piping system. The pressure shall be regulated according to the amount of oil required to be burned. For small variations in the rate of steaming, without changing the number of burners in use, the corresponding change in the quantity of oil supplied shall be made by altering the oil pressure and not by changing the amount of opening of the burners, unless a readjustment is necessary to secure a satisfactory spray.

(b) The air supply: This varies according to the amount of oil to

be burned, and shall be carefully regulated to maintain a steady flame of about the same size from each burner and an amount of smoke that is just visible at the top of the smoke pipe. Too little air will produce excessive smoke and may cause flaming through the slots of the air cones and overheating of the cones themselves. Too much air, while it may prevent smoke, will reduce the efficiency of the boiler. The proper regulation of the velocity with which the air is admitted to the burners is important in connection with preventing panting of the boilers and may be accomplished by varying the opening through the slots in the cones when they are fitted with a movable register for this purpose. When not so fitted, the necessary regulation shall be accomplished by varying the air pressure in the fireroom by means of altering the speed of the fireroom blowers.

(c) The temperature of the oil: The oil is heated to make it thin or fluid enough for efficient spraying at moderate pressures. It is found by experiment that considerable increase in fluidity is obtained by heating the oil to from 150° to 175° F., but above the latter temperature there is no considerable change. If heated much beyond 175° F., there is danger of breaking down the oil and clogging the heaters, strainers, burners, or pipes with the solid particles of carbon thus produced. It is dangerous to heat the oil above its flash point, because if a leak occurs the oil will issue in a flame.

(2) Upon starting fires in a boiler, care shall be taken before lighting fuel-oil burners to insure that the furnace and ash pit are clear of oil and well ventilated, and, in order to avoid a possible back flash, the fireman shall stand well clear of the sight holes and other openings in the furnace front. In lighting burners in addition to those required for raising steam, the oil shall not be turned on until the blowers have been started and the furnace cleared of gas. Similarly, in shutting down, the blowers shall be kept running until all the burners have been shut off. Should a burner become extinguished accidentally, the cause may be due to (a) water mixed with the oil coming from the oil tanks, or from leaky heaters; (b) solid matter choking the burner, due either to fault of the strainers or to carbonizing of the oil in the burner, or (c) water passing over with the oil from the tanks or air chambers on the pumps or oil line. When a burner is choked and cannot be cleared by temporary alterations of the spindle adjustment, it shall be removed at once and thoroughly cleaned. The cleaning shall be very carefully done, care being taken that the outlet holes are not roughened, enlarged, or altered in shape. Burners shall never be left in place disconnected.

(3) When heaters are fitted, special effort shall be made to detect promptly any leaks from the oil to the steam side of the heater. Such leaks allow the oil to pass directly to the boiler water, and, in order to prevent this, the steam pressure on the heater shall be kept higher than the oil pressure, when practicable. At least once during each

watch the drain from the oil heaters shall be tested for the presence of oil, and if oil be found the heater shall be drained and disconnected at once.

(4) When boilers are fitted to burn oil in combination with coal, it shall be borne in mind that the installation is designed to obtain the full power from the boilers when burning coal alone. The oil is provided to make it possible to maintain production of steam on prolonged full-power runs after the coal fires become dirty, or when the trimming of the coal to the firerooms becomes difficult. Therefore, in order to prevent undue forcing when burning both coal and oil, the rate of burning the oil shall not be allowed to exceed fifteen pounds per square foot of grate surface per hour. In burning oil with coal, special attention shall be paid to the opening of the ash-pit doors for regulating the quantity of coal burned. When the fires are clean these doors shall be nearly closed and their opening gradually increased as the fires become dirty. The fires shall be worked so that the grate is well covered, with no holes, and they shall be of modest thickness and even surface. Care shall be taken in handling the fire at the front of the furnace to avoid blocking the air cones and overheating the furnace fronts or cones. The coal shall be fed in small quantities at a time, and the fire doors shall be kept open as short a time as possible. The fire shall be cleaned as would be done when burning coal alone, and, while being cleaned, the burners in that vicinity shall be shut off. The air pressure and the supply of coal and oil shall be carefully regulated, so as to produce the most efficient combustion of both fuels, with a minimum of smoke. Should excess smoke occur, the cause may be (a) fires too heavy, (b) insufficient air pressure in the fireroom or improper opening of air register, or (c) ash-pit doors open too wide, or holes in the fires, thus preventing a sufficient proportion of air passing through the air cones.

3123. Salt Feed.—(1) Any indications of the presence of sea water in the feed tanks, or of considerable increase in salinity of the water in the boilers, shall be investigated promptly and the defect corrected with the least possible delay. As a rule, when sea water becomes unavoidably admitted to the boilers, blowing down shall not be resorted to until the salinity exceeds six hundred grains of chlorine per gallon in Babcock and Wilcox or fire-tube boilers, three hundred grains in Niclausse boilers, and one hundred and fifty grains in small bent tube or torpedo boat boilers. When bottom blow valves are used to reduce the salinity of boilers under steam, they shall be quickly opened wide for a few seconds only, and at intervals of not less than four hours. Too frequent use of these valves shall be avoided, as it causes them to leak, indirectly wastes fuel, and promotes corrosion.

Emptying Boilers.—(2) When it is required to empty the boilers, they shall not be blown down, as this practice is liable to cause leaky tubes

and joints, but the water shall be allowed to remain until it becomes cool, and then shall be either pumped or drained out.

3124. Evaporators.—As the efficient operation of the evaporators depends upon varying conditions of steam pressure, height and density of water, etc., according to the state of the heating surfaces at any time, constant attention shall be given to secure the best working conditions at all times, in order to obtain the maximum production of water at the most economical rate of expenditure of fuel. Evaporators shall be operated in double effect when so fitted. The density of the water shall not ordinarily be allowed to exceed $3/32$ by the salinometer, otherwise there is danger of hydrochloric acid being produced by decomposition of magnesium chloride, which, passing to the distillers, will render the fresh water acid. The fresh water distilled shall be tested at frequent and regular intervals, in order to insure that it is being produced in a pure state and free from salt.

3125. Oil in Internal-Combustion Engines.—In internal combustion engines designed for splash lubrication in the crank cases, care shall be taken not to carry the level of the lubricating oil too high. If the level is kept much higher than is necessary for proper lubrication, the oil is liable to pass the pistons and burn in the combustion spaces, not only fouling the cylinders and clogging the valves and piston rings but resulting in unnecessary waste of the lubricating oil.

3126. Pneumatic System.—(1) In starting an air compressor, it shall be run slowly until it becomes warm and until there is a pressure of at least twenty-five pounds in the accumulator to provide an air cushion. To avoid overheating of the air cylinder, compressors of the Westinghouse or similar type, not fitted with water jackets, shall not be run at too high a rate of speed, and care shall be taken that the governor acts properly.

(2) After using air compressing machinery for any purpose, great care shall be taken to see that the engines, pumps, separators, charging columns, and reservoirs are blown out and thoroughly drained of water. The pressure gauges shall be left open to the spaces to which they are attached, so that any pressure that might remain in the system will be indicated, in order to avoid accidents upon disconnecting any parts.

(3) A spare set of leather cup washers shall always be kept ready for use, such leathers being kept in tins filled with neat's-foot, castor, or other oil suitable for keeping them soft and in proper condition.

(4) The oil used for lubricating the internal parts of the air cylinders when the packings are of leather shall be neat's-foot, or, if that is not obtainable, castor oil or other suitable oil shall be used. Such oil, however, shall not be allowed to enter any steam cylinder.

3127. Preparation for Collision.—(1) Before going into action, all articles which might be displaced by a collision shall be secured or so

disposed that no injury thereby can be done to the machinery or to any person.

(2) When intending to ram, or likely to be rammed, notice shall be given from deck to the engine room by any prompt method of communication, so that men, tools, etc., may not be thrown down or against moving parts of the machinery.

(3) To prevent the passage of water from the boilers into the engines when collision takes place, the separators shall be blown out and their drains left open until the engines are safely reversed.

(4) In ramming, being rammed, and generally in cases of collision, men shall go to their stations for starting all bilge pumps and other means provided for freeing the ship of water in case of dangerous leakage.

3128. Receipt and Stowage of Coal.—(1) Coal shall not be taken on board wet if it can be avoided, and care shall be taken to keep it dry in the bunkers, as moisture sometimes causes a rapid and dangerous generation of heat and gas, resulting in spontaneous combustion. Before the decks are washed down after coaling, the solid bunker plates shall be replaced and made tight to prevent water getting into the bunkers. Should there be any indication of spontaneous combustion, it shall at once be reported to the officer of the deck and to the executive and commanding officers.

(2) The stowage of coal in the bunkers shall be limited to the height of the lower edge of the beams, in order that sufficient vacant space above the surface of the coal shall be preserved for proper ventilation. The capacities of the bunkers shall be calculated on this basis.

(3) The ventilation pipes fitted to the bunkers shall be kept clear, and they shall always be kept open for ventilation, except when running the blowers and a loss of air pressure in the firerooms through open bunkers would be caused thereby. The plates of all fixed coaling trunks and coal bunkers not provided with permanent ventilation fittings shall be taken off periodically to ventilate these spaces. This shall be done at frequent intervals after coaling, as the evolution of gas owing to the breaking up of the coal is very rapid during and for some days after the operation of coaling ship. It shall be borne in mind that to secure efficient ventilation there must be at least two openings, one for the admission of pure air and another for the escape of foul air; and, where permanent ventilation fittings do not include both, the bunker plates shall be taken off periodically as required above. Care shall be taken to thoroughly ventilate such bunkers before any men are sent to work in them.

(4) No open light shall be permitted in a coal bunker, or within twenty feet of an opening into a coal bunker, until the bunker has been thoroughly ventilated and it has been ascertained by a safety lamp or other suitable means that it does not contain explosive gas. Special

precautions in this respect shall be taken for a few days after coaling. In any case in which the distance of twenty feet is impracticable the distance kept shall be as great as possible.

(5) During the intervals between steaming periods, and at other times when it may be done to advantage, the coal shall be trimmed from the upper and more remote bunkers into close proximity to the bunker doors of the firerooms where it will eventually be required for use. This is specially important preparatory to steaming at a high rate of speed, when a considerable supply of coal will be needed. The engineer officer shall keep himself informed of the general distribution of coal in the bunkers.

(6) Coal shall not be stowed in the firerooms in such quantities as to interfere with working the boilers or to cover up the handles or wheels of valves, or to get into the bilges, thereby possibly choking the pump suction and strainers and endangering the safety of the ship.

B

LIST OF SPARE PARTS USUALLY CARRIED BY U. S. BATTLESHIPS AND ARMORED CRUISERS.

SPARE PARTS FOR MAIN ENGINES.

Parts.	Number to be Furnished.
Piston rods	1 of each size and pattern.
Valve stems	Ditto.
Crosshead slippers	Ditto.
Metallic packing, piston rods	All for both engines.
Metallic packing, valve stems	Ditto.
Eccentric straps, bolts, and liners complete	1 set of each pattern.
Crank-pin, brasses, top	1 of each pattern.
Crank-pin brasses, bottom	Ditto.
Liners for crank-pin brasses	2 of each pattern.
Bolts for crank-pin brasses	Ditto.
Crosshead-pin brasses, top	Ditto.
Crosshead-pin brasses, bottom	Ditto.
Liners for crosshead-pin brasses	4 of each pattern.
Bolts for crosshead-pin brasses	Ditto.
Eccentric-rod brasses, top	2 of each pattern.
Eccentric-rod brasses, bottom	Ditto.
Liners for eccentric-rod brasses	4 of each pattern.
Main-bearing brasses, top, if fitted	1 of each pattern.
Main-bearing brasses, bottom	Ditto.
Bolts for main bearing	4 of each pattern.
Valve-gear brasses, bushings, etc., not specifically mentioned above	1 of each size and pattern.
Piston rings, complete	1 for each piston.
Piston-ring springs	All for each piston.
Piston-follower bolts and nuts	25 of each size and pattern.
Cylinder-head bolts	Ditto.
Coupling bolts and nuts	3 of each size and pattern.
Split pins, special	Ditto.

BOILER SPARES.

Parts.	Quantity to be Furnished.
Grate bars	Complete for 3 boilers.
Bearers	Ditto.
Dead plates	Ditto.
Furnace-door lintels, complete.....	Ditto.
Furnace-door linings	Ditto.
Furnace-door jambs, right end, left end, and center, complete	Ditto.
Flame plates, complete	Complete for 1 boiler.
Oil fuel burners, complete	20 per cent.
Casing for oil fuel burners, with doors.....	Ditto.
Ash-pit doors	Complete for 1 boiler.
Manhole plates, with bolts, nuts and dogs...	Complete for 2 boilers.
Gauge cocks	Ditto.
Water gauges, complete	Ditto.
Water-gauge glasses	500 per cent.
Feed check valves	2 to each hand.
Surface blow valves	1 to each hand.
Bottom blow valves	Ditto.
Handhole plates, including superheater, with bolts, nuts, and dogs	5 per cent of all.
Special fire brick	10 per cent of all.
Special fittings	As may be directed.
All tubes of each size and shape, including side and cross boxes, if fitted, and superheater	5 per cent.
Springs for boiler safety valves	Complete for 1 boiler.

MISCELLANEOUS SPARE PARTS.

Parts.	Quantity to be Furnished.
Spring-bearing brasses	1 top and bottom brass for each size fitted.
Condenser tubes of each length	5 per cent.
Feed-water heater tubes	10 per cent.
Distiller tubes	25 per cent.
Evaporator tubes	5 per cent.
Condenser-tube glands	10 per cent.
Oil-heater tubes	Enough for 1 heater.
Springs for all safety valves except boiler safety valves	1 of each size, pressure and type.
Springs for all relief valves	Ditto.
Springs for all reducing valves	Ditto.
Diaphragms for reducing valves	2 of each size, pressure and type.
Removable seats for all gate valves	Ditto.
Propeller-blade bolts	100 per cent.
Zincs not made from standard plates	Ditto.
Thrust-bearing horseshoes	All for either bearing.
Baskets for bilge strainers	100 per cent.
Baskets for oil strainers	Ditto.
Collar bolts for all purposes	10 per cent.
Cutters for turbine tube cleaners	600 per cent.

SPARE PARTS FOR AUXILIARY ENGINES.

Parts.	Ash-hoist engines.	Circulat- ing pump engines.	Turning engines.	Reversing engines.
Connecting rods, of each size and pattern fitted	1	1	..	..
Eccentric rods, with straps, bolts and nuts, of each size and pattern fitted	1	1	1	..
Piston rods and nuts, complete, of each size and type fitted	3	1	1	..
Crosshead and slippers, complete.	1	1	1	..
Valve stems and nuts, complete, of each size and pattern fitted	3	1	1	..
Brasses and bushings, for each one fitted	1	1	1	1
Bolts and nuts of crank pin and cross-head brasses	50%	50%	50%	..
Metallic packing	50%	50%	50%	..
Packing rings for pistons and piston valves, for each size and pattern fitted, complete	2	1	1	1
Piston springs, studs and nuts, complete, for pistons	Enough for one piston of each size and pattern.			

Typical Installation of Workshop Machinery on a Battleship.

There will be provided and installed in the general workshop the following machine tools, viz.:

One screw-cutting, back-geared, extension-gap lathe, to swing at least 28 inches over the upper and 48 inches over the lower ways, with not less than 10 feet between the centers when extended.

One tool-room lathe, 14-inch swing, taking 18 inches between centers.

One screw-cutting, back-geared tool-room lathe, to swing at least 14 inches over the ways and not less than 4 feet between the centers.

One column tool-room shaper, of at least 15-inch stroke and not less than 15-inch traverse.

One 30-inch radial drill, to drill up to 1½ inches diameter at maximum position of spindle and arm, spindle fitted for No. 4 Morse taper, with requisite number of sockets.

One 16-inch high-speed sensitive drill.

One universal milling machine with overhanging arm to have at least 18-inch longitudinal feed of table, 13-inch vertical movement, and 4½-inch traverse. Spindle to be fitted for No. 9 Brown and Sharpe taper.

One double emery grinder on column, with attachments for surface grinding and hoods over grinding wheels.

One portable cylinder-boring machine, with latest attachments, for boring cylinders in place.

Six machinist's swivel-bottom vises, with jaws at least $5\frac{1}{4}$ inches wide, opening 8 inches, with pipe jaws and copper vise lips.

One approved steel blacksmith's forge, of the portable folding type, with all the necessary tools and fixtures. The blast wheel will be about 12 inches diameter, turned by a crank. The gearing enclosed in a dust-proof case. When set up, the pan will be about 24 inches square. The pan and hood will form a case which, when the forge is dismantled and the parts placed inside, will be a strong, compact steel box, provided with handles for carrying. This forge will be supplied with a 120-pound portable anvil, with block, and a set of tools of assorted sizes.

There will be provided and installed in the blacksmith space an approved blacksmith's forge, about 42 inches square by about 25 inches high. Forge will be complete with electrically driven blast fan, tuyere, blast gate, pipe connecting as directed, etc., and with a set of tools of assorted sizes.

One steel blacksmith's anvil of 180 pounds weight, with block, secured in position, face $15\frac{1}{2}$ inches by $4\frac{1}{4}$ inches, horn 10 inches long, with a cutter hole 1 inch square.

All tools will be of the best design, material, quality, and workmanship, will have the latest improvements and all necessary attachments and wrenches, and, when required, provision will be made for working them by hand.

The lathes will have a complete set of change gears for cutting United States Navy standard screw threads, including $11\frac{1}{2}$ threads per inch.

When submitted for approval, there must be furnished full descriptions, cuts, or drawings showing details of the machine; also the name of the maker, the net weight, the space occupied, and a list of the tools and attachments furnished.

All machines will be furnished complete with a full set of cutting tools and all necessary attachments for miscellaneous work.

The machine tools will each be driven by an independent inclosed variable-speed electric motor or equivalent variable-speed control, as approved, with inclosed controlling panels located where directed. The lathes will be provided with reversible drum-type controllers attached to the carriage. All motors will be direct connected without the intervention of belting.

FOUNDRIY.

The foundry will be equipped with the following outfit, all as approved:

One oil-burning crucible furnace of bureau standard type.

Five No. 35 crucibles.

Three No. 150 crucibles.

One pair No. 35 lifting tongs.

One pair No. 150 lifting tongs.

- One No. 35 pouring shank, one end single.
- One No. 150 pouring shank, both ends double.
- One 5-foot fire tool.
- One No. 2 molder's shovel.
- Two hard-wood bench rammers, 14 inches long by $3\frac{1}{2}$ inches diameter of butts.
- One 18-inch riddle, double wood rim, No. 4 mesh brass wire.
- One 18-inch riddle, double wood rim, 4 supporting wires, No. 8 mesh brass wire.
- One 18-inch riddle, double wood rim, 4 supporting wires, No. 12 brass wire.
- One pair 10-inch calfskin molder's bellows.
- One combined pattern and bench brush, tampico and bristle.
- One 1-inch flat brush, camel's hair, brass bound.
- One $1\frac{1}{2}$ -inch by 6-inch square end trowel.
- One $1\frac{1}{4}$ -inch slick and oval spoon.
- One $\frac{5}{8}$ -inch Yankee heel and flat slick.
- One $\frac{5}{8}$ -inch flange and bead slick.
- One 2-inch corner slick.
- One $1\frac{1}{2}$ -inch pipe slick.
- One $\frac{1}{4}$ -inch by 14-inch lifter.
- One $\frac{1}{2}$ -inch T-head bench lifter with bent slick.
- Two 12-inch by 16-inch by 16-inch interchangeable cast-iron flasks.
- Two 18-inch by 24-inch by 10-inch interchangeable cast-iron flasks.
- Work bench, sand trough, scrap-metal bins, tool locker, racks, fuel-oil tank, etc.

C

TABLE XIV.
HYPERBOLIC LOGARITHMS.

$$\log_e N = 2.302585 \times \log_{10} N.$$

No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.
1.1	0.0953	3.7	1.3083	6.3	1.8405	8.9	2.1861
1.2	0.1823	3.8	1.3350	6.4	1.8563	9.0	2.1972
1.3	0.2624	3.9	1.3610	6.5	1.8718	9.1	2.2083
1.4	0.3365	4.0	1.3863	6.6	1.8871	9.2	2.2192
1.5	0.4055	4.1	1.4110	6.7	1.9021	9.3	2.2300
1.6	0.4700	4.2	1.4351	6.8	1.9169	9.4	2.2407
1.7	0.5306	4.3	1.4586	6.9	1.9315	9.5	2.2513
1.8	0.5878	4.4	1.4816	7.0	1.9459	9.6	2.2618
1.9	0.6419	4.5	1.5041	7.1	1.9601	9.7	2.2721
2.0	0.6931	4.6	1.5261	7.2	1.9741	9.8	2.2824
2.1	0.7419	4.7	1.5476	7.3	1.9879	9.9	2.2925
2.2	0.7885	4.8	1.5686	7.4	2.0015	10.0	2.3026
2.3	0.8329	4.9	1.5896	7.5	2.0149	10.5	2.3513
2.4	0.8755	5.0	1.6094	7.6	2.0281	11.0	2.3979
2.5	0.9163	5.1	1.6292	7.7	2.0412	11.5	2.4430
2.6	0.9555	5.2	1.6487	7.8	2.0541	12.0	2.4849
2.7	0.9933	5.3	1.6677	7.9	2.0669	12.5	2.5262
2.8	1.0296	5.4	1.6864	8.0	2.0794	13.0	2.5649
2.9	1.0647	5.5	1.7047	8.1	2.0919	13.5	2.6027
3.0	1.0986	5.6	1.7228	8.2	2.1041	14.0	2.6391
3.1	1.1314	5.7	1.7405	8.3	2.1163	15.0	2.7081
3.2	1.1632	5.8	1.7579	8.4	2.1282	16.0	2.7726
3.3	1.1939	5.9	1.7750	8.5	2.1401	17.0	2.8332
3.4	1.2238	6.0	1.7918	8.6	2.1518	18.0	2.8904
3.5	1.2528	6.1	1.8083	8.7	2.1633	19.0	2.9444
3.6	1.2809	6.2	1.8245	8.8	2.1748	20.0	2.9957

D

TABLE XV.

SATURATED-STEAM TABLES.*

Absolute pressure, pounds per sq. inch.	Temperature of boiling-point, degrees F.	Heat of the liquid from 32° F.	Total heat from 32° F.	Latent heat.	Heat to overcome internal resistance.	Heat to overcome external resistance.	Thermal units actually contained in the steam above 32° F.	Weight of a cub. ft. in pounds.	Cub. ft. per pound.
<i>p</i>	<i>t</i>	<i>h</i>	<i>H</i>	<i>L</i>	<i>p</i>	<i>E</i>	<i>I = H - E</i>	<i>w</i>	<i>V</i>
1.0	102.0	70.0	1113.1	1043.0	981.1	61.9	1051.2	0.00299	334.6
2.0	126.3	94.4	1120.5	1026.1	961.9	64.2	1056.3	0.00576	173.6
3.0	141.6	109.8	1125.1	1015.3	949.5	65.8	1059.3	0.00844	118.4
4.0	153.1	121.4	1128.6	1007.2	940.4	66.8	1061.8	0.01107	90.31
5.0	162.3	130.7	1131.5	1000.8	933.1	67.7	1063.8	0.01366	73.22
6.0	170.1	138.6	1133.8	995.2	926.7	68.5	1065.3	0.01622	61.67
7.0	176.9	145.4	1135.9	990.5	921.4	69.1	1066.8	0.01874	53.37
8.0	182.9	151.5	1137.7	986.2	916.5	69.7	1068.0	0.02112	47.07
9.0	188.3	156.9	1139.4	982.5	912.4	70.1	1069.3	0.02374	42.13
10.0	193.2	161.9	1140.9	979.0	908.4	70.6	1070.3	0.02621	38.16
11.0	197.8	166.5	1142.3	975.8	904.8	71.0	1071.3	0.02866	34.88
12.0	202.0	170.7	1143.6	972.9	901.5	71.4	1072.2	0.03111	32.14
13.0	205.9	174.6	1144.7	970.1	898.4	71.7	1073.0	0.03355	29.82
14.0	209.6	178.3	1145.8	967.5	895.5	72.0	1073.8	0.03600	27.79
14.7	212.0	180.7	1146.6	965.8	893.5	72.3	1074.2	0.03758	26.64
15.0	213.0	181.8	1146.9	965.1	892.6	72.5	1074.4	0.03826	26.15
16.0	216.3	185.1	1147.9	962.8	890.0	72.8	1075.1	0.04067	24.59
17.0	219.4	188.3	1148.9	960.6	887.6	73.0	1075.9	0.04307	23.22
18.0	222.4	191.3	1149.8	958.5	885.3	73.2	1076.6	0.04547	22.00
19.0	225.2	194.1	1150.7	956.6	883.2	73.4	1077.3	0.04786	20.90
20.0	227.9	196.9	1151.5	954.6	881.0	73.6	1077.9	0.05023	19.91
21.0	230.5	199.5	1152.3	952.8	879.0	73.8	1078.5	0.05259	19.01
22.0	233.1	202.0	1153.0	951.0	877.0	74.0	1079.0	0.05495	18.20
23.0	235.5	204.5	1153.7	949.2	875.0	74.2	1079.5	0.05731	17.45
24.0	237.8	206.8	1154.4	947.6	863.2	74.4	1080.0	0.05966	16.76
25.0	240.0	209.1	1155.1	946.0	871.5	74.5	1080.6	0.06199	16.13
26.0	242.2	211.2	1155.8	944.6	869.9	74.7	1081.1	0.06432	15.55
27.0	244.3	213.4	1156.5	943.1	868.2	74.9	1081.6	0.06666	15.00

* The above steam data are taken from Prof. Peabody's valuable "Saturated Steam Tables," by kind permission of the author.

TABLE XV—Continued.

Absolute pressure, pounds per sq. inch.	Temperature of boiling-point, degrees F.	Heat of the liquid from 32° F.	Total heat from 32° F.	Latent heat.	Heat to overcome internal resistance.	Heat to overcome external resistance.	Thermal units actually contained in the steam above 32° F.	Weight of a cub. ft. in pounds.	Cub. ft. per pound.
<i>p</i>	<i>t</i>	<i>h</i>	<i>H</i>	<i>L</i>	<i>p</i>	<i>E</i>	<i>I = H - E</i>	<i>w</i>	<i>V</i>
28.0	246.4	215.4	1157.1	941.7	866.7	75.0	1082.1	0.06899	14.49
29.0	248.3	217.4	1157.7	940.3	865.1	75.2	1082.5	0.07130	14.03
30.0	250.3	219.4	1158.3	938.9	863.6	75.3	1083.0	0.07360	13.59
31.0	252.1	221.3	1158.8	937.5	862.0	75.5	1083.3	0.07590	13.18
32.0	254.0	223.1	1159.4	936.3	860.7	75.6	1083.8	0.07821	12.78
33.0	255.8	224.9	1159.9	935.0	859.2	75.8	1084.1	0.08051	12.41
34.0	257.5	226.7	1160.4	933.7	857.8	75.9	1084.5	0.08280	12.07
35.0	259.2	228.4	1161.0	932.6	856.6	76.0	1085.0	0.08508	11.75
40.0	267.1	236.4	1163.4	927.0	850.3	76.7	1086.7	0.09644	10.37
45.0	274.3	243.6	1165.6	922.0	844.8	77.2	1088.4	0.1077	9.287
50.0	280.8	250.2	1167.6	917.4	839.7	77.7	1089.9	0.1188	8.414
55.0	286.9	256.3	1169.4	913.1	834.9	78.2	1091.2	0.1299	7.696
60.0	292.5	261.9	1171.2	909.3	830.7	78.6	1092.6	0.1409	7.096
65.0	297.8	267.2	1172.7	905.5	826.5	79.0	1093.7	0.1519	6.583
70.0	302.7	272.2	1174.3	902.1	822.7	79.4	1094.9	0.1628	6.144
75.0	307.4	276.9	1175.7	898.8	819.1	79.7	1096.0	0.1736	5.762
80.0	311.8	281.4	1177.0	895.6	815.5	80.1	1096.9	0.1843	5.425
85.0	316.0	285.8	1178.3	892.5	812.1	80.4	1097.9	0.1951	5.125
90.0	320.0	290.0	1179.6	889.6	808.9	80.7	1098.9	0.2058	4.858
95.0	323.9	294.0	1180.7	886.7	805.8	80.9	1099.8	0.2165	4.619
100.0	327.6	297.9	1181.9	884.0	802.8	81.2	1100.7	0.2271	4.403
105.0	331.1	301.6	1182.9	881.3	799.9	81.4	1101.5	0.2378	4.206
110.0	334.6	305.2	1184.0	878.8	797.1	81.7	1102.3	0.2484	4.026
115.0	337.9	308.7	1185.0	876.3	794.4	81.9	1103.1	0.2589	3.862
120.0	341.0	312.0	1186.0	874.0	791.9	82.1	1103.9	0.2695	3.711
125.0	344.1	315.2	1186.9	871.7	789.4	82.3	1104.6	0.2800	3.572
130.0	347.1	318.4	1187.8	869.4	786.9	82.5	1105.3	0.2904	3.444
135.0	350.0	321.4	1188.7	867.3	784.7	82.6	1106.1	0.3009	3.323
140.0	352.8	324.4	1189.5	865.1	782.3	82.8	1106.7	0.3113	3.212
145.0	355.6	327.2	1190.4	863.2	780.2	83.0	1107.4	0.3218	3.107
150.0	358.3	330.0	1191.2	861.2	778.1	83.1	1108.1	0.3321	3.011
155.0	360.9	332.7	1192.0	859.3	776.0	83.3	1108.7	0.3426	2.919
160.0	363.4	335.4	1192.8	857.4	774.0	83.4	1109.4	0.3530	2.833
165.0	365.9	338.0	1193.6	855.6	772.0	83.6	1110.0	0.3635	2.751
170.0	368.3	340.5	1194.3	853.8	770.1	83.7	1110.6	0.3737	2.676
175.0	370.6	343.0	1195.0	852.0	768.2	83.8	1111.2	0.3841	2.603
180.0	373.0	345.4	1195.7	850.3	766.4	83.9	1111.8	0.3945	2.535
185.0	375.23	347.8	1196.4	848.6	764.6	84.0	1112.4	0.4049	2.470
190.0	377.4	350.1	1197.1	847.0	762.9	84.1	1113.0	0.4153	2.408
195.0	379.6	352.4	1197.7	845.3	761.1	84.2	1113.5	0.4257	2.349
200.0	381.7	354.6	1198.4	843.8	759.5	84.3	1114.1	0.4359	2.294
250.0	401.0	374.7	1204.2	829.5	744.5	85.0	1119.2	0.5393	1.854
300.0	417.4	391.9	1209.3	817.4	732.0	85.4	1123.9	0.6440	1.554
400.0	444.9	419.8	1217.7	797.9	712.3	86.2	1131.5	0.8572	1.167

E

TABLE XVI.

TABLE OF MEAN PRESSURE RATIOS (Northcott).

To find the mean pressure exerted throughout the stroke, multiply the initial pressure by the number opposite the ratio of expansion, in the column corresponding with the conditions of expansion.

$r = \frac{v_2}{v_1}$ ratio of expansion.	$\frac{p_m}{p_1} = \frac{1 + \log_e r}{r}$ hyperbolic curve.	$\frac{p_m}{p_1} = \frac{17 - 16r - \frac{1}{18}}{r}$ saturation curve.	$\frac{p_m}{p_1} = \frac{10 - 9r - \frac{1}{9}}{r}$ adiabatic curve.
1.0	1.000	1.000	1.000
1.5	.934	.931	.928
2.0	.846	.840	.833
2.5	.766	.756	.748
3.0	.699	.688	.680
3.5	.642	.631	.622
4.0	.596	.583	.572
4.5	.555	.542	.532
5.0	.522	.506	.496
5.5	.492	.477	.467
6.0	.465	.450	.440
7.0	.421	.405	.393
8.0	.385	.370	.356
9.0	.355	.340	.327
10.0	.330	.314	.303
11.0	.309	.294	.283
12.0	.290	.275	.264
13.0	.274	.259	.248
14.0	.260	.245	.234
15.0	.247	.232	.221
16.0	.236	.221	.211
17.0	.226	.211	.201
18.0	.216	.202	.192
19.0	.208	.193	.183
20.0	.200	.186	.177
21.0	.192	.178	.169
22.0	.186	.172	.163
23.0	.180	.167	.158
24.0	.174	.160	.151
25.0	.169	.155	.146

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SPECIFICATIONS FOR INSPECTION OF MATERIAL.

PART I.

Steel and Iron for Use in the Construction of Machinery Coming Under the Cognizance of the Bureau of Steam Engineering, Navy Department.

(Revised 1911.)

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GENERAL INSTRUCTIONS FOR THE INSPECTION OF STEEL AND IRON MATERIAL.

Coming Under the Cognizance of the Bureau of Steam Engineering,
Navy Department.

OFFICE AND INSPECTORS.

1. Access to Work and Information.—The department shall have the right to keep inspectors at the works, who shall have free access at all times to all parts thereof and be permitted to examine the raw material and to witness the processes of manufacture.

Contractors and manufacturers shall furnish all the information and facilities the inspector may require for proper inspection under these specifications.

2. Inspector's Office and Furniture.—Each firm manufacturing material shall furnish the inspectors, free of expense, with such facilities as may be necessary for the proper transaction of their business as agents of the Government.

EXPENSE.

3. Handling Material.—All handling of material necessary for purposes of inspection shall be done at the expense of the contractor.

4. Making Tests.—All test specimens necessary for the determination of the qualities of material used shall be prepared and tested at the expense of the contractor.

REJECTION AFTER LEAVING MANUFACTURER.

5. **Rejection at Destination.**—Material may be rejected at a navy yard or other place of delivery for surface or other defects either existing on arrival or developed in working or storage, even though such material may have previously passed inspection by the Government inspector at the place of manufacture. In such cases the contractor must make good any material rejected.

INSTRUCTIONS FOR INSPECTORS.

6. **Notification to Manufacturers.**—Upon receipt of instructions from the Bureau of Steam Engineering or of a request from an inspector of machinery, another inspector of material, a representative of another bureau, or a navy pay office to inspect an order for material, the inspector will at once inform the manufacturers of this fact, and also that the order must not be shipped until after it has passed Government inspection, without the inspector's consent. He will also direct the manufacturer to notify him when the material will be ready for inspection.

7. **Inspection During Process of Manufacture.**—The inspector should keep in touch with the work throughout its manufacture, and make such efforts as are practicable to secure delivery within contract time.

8. **Inspection of Material Without Instructions.**—Any material which a manufacturer may present to the naval inspector will be inspected, provided it is without doubt material intended to be inspected by the Bureau of Steam Engineering, and also provided there is no doubt that the specifications on the company's orders are in accordance with the usual specifications for that class of material, and that the material ordered is covered by approved plans for its use. Immediate report to the bureau will be made in such cases.

9. **Notifying Bureau of Manufacture of Finished Material Without Orders for Inspection.**—The Bureau of Steam Engineering desires to be notified by the naval inspector of all Government material being manufactured in his district for the Bureau of Steam Engineering upon which no inspection has been ordered. The bureau will then determine whether or not inspection is necessary, if not requested, after the following requirements have been complied with:

Inspectors will communicate directly with inspectors of machinery, other inspectors of engineering material, representatives of other bureaus, navy pay offices, and commandants of navy yards in regard to material that is under inspection, or material the inspection of which has not been requested.

Inspectors of engineering material requesting inspection of material by another inspector will forward the contractor's order in duplicate, with signed certificate on face that material ordered is in accordance with approved plans and specifications, to the inspector of material in the district where the material ordered is to be manufactured. A mani-

fold copy of this order will be forwarded to the bureau. Inspectors requesting information in regard to the inspection of material from a navy pay officer will communicate direct with the purchasing pay officer, requesting that duplicate copies of order be forwarded and that a copy sent direct to the Bureau of Steam Engineering. A carbon copy of this letter to the purchasing pay officer will be forwarded to the bureau.

ORDERS, LISTS AND INVOICES.

10. Contractor or Subcontractor will supply the inspector of engineering material or inspector of machinery with the following:

(a) Quadruplicate copies of all orders placed with manufacturers or subcontractors, stating, when possible, the purpose of each item ordered and the specifications for same. On all copies the inspector will certify that the material is in accordance with approved plans and specifications. One copy will be retained by the inspector, two copies will be sent to the inspector who will inspect the material, and one copy forwarded to the bureau. The copy forwarded to the bureau shall be without letter or endorsement slip. Blue prints of parts will not be forwarded to the bureau unless they give data of the order. The order must also state explicitly what treatment, if any, other than machining, is to be given after leaving the manufacturer's works.

(b) Triplicate blueprints or sketches when such are necessary in placing orders.

When possible, a separate order should be placed for each vessel or contract.

The contractors and manufacturers will save considerable time and correspondence by referring directly to the inspector all matters relating to inspection of material.

11. Manufacturer Will Furnish the Inspector—

1. A list of all ingots in each heat, giving—

(a) Number of heat.

(b) Weight of each ingot.

(c) Information as to marking and arranging of ingots. (The marking and arranging to be such as to make identification easy.)

(d) The amount of discard at top and bottom of ingots (when required by the inspector).

(e) The number and heights of the heads and risers (when required by the inspector).

2. Copies of mill orders.

3. Copy of manufacturer's chemical analysis of material when possible.

12. Acceptance and Shipment of Material—

(a) No material will be shipped by a manufacturer or sub-contractor except by direction of the inspector or other naval authority.

(b) The manufacturer will furnish the inspector, immediately after each shipment of material, an invoice of shipment in quadruplicate, which must give—

1. Identification marks.
2. Dimensions of each object.
3. Actual weights of each object or of each lot of similar objects of like dimensions. When material is ordered by weight or gauge, the estimated weights must also be given.

4. Date of shipment.

5. Names of consignor and consignee.

6. Number and initials of car when shipment is made in carload lots.

7. Number of pieces and packages shipped.

(c) Any delay in furnishing this information will cause a like delay in the acceptance of the material at destination and in the preparation of vouchers incident to payments for the same.

(d) The manufacturers will show the inspector all schedules and orders of material for the Government placed with them by the contractors.

13. The Inspector will Forward a Copy of Each Shipping Report to the Bureau of Steam Engineering, and also to the naval inspector at the place to which the material is shipped. If material is intended for a navy yard or naval station, the inspector will also forward two copies of each shipping report to the general storekeeper. If the material inspected belongs to another bureau, one copy of shipping report will also be sent to the bureau concerned. Shipping reports will give—

(a) The information required by printed form.

(b) The stamp or stamps which appear on the material inspected, on the casing containing the material, or on tags on car in which the material is shipped, and on what part of material, box, or car the stamp is placed.

(c) Date that the manufacturer's report was received, stating when the material was or would be ready for inspection.

(d) Dates on which the material was inspected.

(e) Date of authorization of shipment.

(f) Initials of assistant inspector who made the inspection.

(g) Certificate that copies of the report have been sent to various officials.

(h) Whether the shipment completes the order.

(i) In the case of a foreign shipment it will be stated that the inspector has verified the packing and that the packing has been sealed.

(j) Information of special treatment or inspection.

(k) File number of office copy.

The date of report will be that of date of shipment. Where the manufacturers furnish their own form of report, this may be pasted onto the regular form, but all information required will be furnished (but this practice should be followed only in exceptional cases).

A shipping report will only cover material supplied on one order. Final report will state the number of shipments made on account of the order.

The reports going to the naval official at destination must bear the autographic signature of the inspector, but the other copies may be signed with a stamp.

STAMPS.

14. Identification of Test Coupons and Material from Which Taken.—The coupons, castings, parts of sheet metal, or other material from which tests or analyses are to be made, must be stamped with the U S or inspector's personal stamp for identification. This stamp will not be removed by the manufacturer during any part of finishing process except after duplication by a Government inspector.

The removal of any Government stamp from material without authority of the inspector will be sufficient reason for rejection of that material.

15. Anchor Stamp.—The anchor stamp shall not be put on any material until it has been inspected, weighed, and passed, ready for shipment, and must always be considered as evidence of final inspection.

16. Identification Stamps.—The U S stamp, or the inspector's personal stamp, will be used when necessary for identification during the progress of work.

Each object which has passed inspection shall be clearly marked with the necessary stamps, and these stamps on large, rough work shall be encircled by a ring of paint.

Ingots, blooms, and other such material intended to be cut up shall have the stamps above mentioned put on in three places, viz., near each end and near the middle, and encircled by paint marks.

17. Stamps on Boxes.—In the case of small articles passed and packed in bulk, or in the case of material which would be injured by stamping, the above-mentioned stamps shall be applied to the boxing or packing material of the articles, instead of to the articles themselves.

18. Carload Lots—Tags.—If the material is shipped in a box car containing no other freight, the car must be sealed and the anchor stamp and the inspector's personal stamp placed on the seal as well as on a tag on the inside of the car near the door.

19. Nonacceptance of Material Without Stamps.—No material will be received at a navy yard, naval station, or building yard for incorporation into naval vessels unless it bears either upon its surface or that of its packing the stamps noted on the shipping report as evidence that it has passed the required government inspection.

20. Care of Stamps.—Stamps must be carefully guarded by the inspectors and assistant inspectors to whom they may be intrusted, to prevent their use by unauthorized persons.

GENERAL QUALITY.

21. General Character of Material.—All material shall be of domestic manufacture, of uniform quality throughout the mass of each object, and free from all defects.

22. Uniform Quality to be Insured.—The discarding of inferior portions of ingots, the fabrication, treatment, and manufacture generally, shall be so conducted as to insure uniformity in the quality of the metal of each heat, lot, or object submitted for inspection.

23. Special Material or Special Treatment.—With the approval of the Bureau of Steam Engineering, special material or special treatment, or both, may be used to obtain the qualities specified.

ANNEALING AND TREATMENT.

24. Annealing Method.—The whole of an object to be annealed shall be subjected to the same degree of heat at the same time. All annealing shall be done at a proper heat. The number of hours requisite for raising the object to a proper temperature, the length of time during which it shall be soaked at that temperature, and the period required for slow cooling in the furnace may be prescribed by the Bureau of Steam Engineering. (See article 23.)

25. Treatment of Lots.—All objects tested as a heat shall have received the same treatment.

GENERAL TEST REQUIREMENTS.

26. Tests and Acceptance.—All material for which tests are prescribed shall be inspected and tested by a naval inspector and passed by him, subject to restrictions mentioned herein, before acceptance by the Navy Department.

27. Treatment of Test Pieces.—Test pieces shall not be subjected to any treatment or process except machining to size; and such pieces shall not be cut off until the plate or object shall have received final treatment and been stamped by the inspector, except in those special cases mentioned in the following specifications.

28. Flaws in Test Pieces.—Test pieces which show defective machining or which after breaking show flaws, or which break outside of the measuring points, may be discarded, and the inspector will select others in their stead.

29. Test Pieces for Lots.—Test pieces which represent heats or lots shall be taken, as nearly as the case will permit, so as to represent the metal which was nearest to the top of the ingot.

30. Location of Test Pieces.—All test pieces of forgings, and of rolled bars which are too large to be pulled in their full sizes shall, unless otherwise specified, be taken at a distance from the longitudinal axis of the object equal to one-quarter of the greatest transverse dimension of the body of the object, not including palms and flanges.

The test pieces should be taken from a part of the material which,

with the exception of palms or flanges, has not been reduced by forging or rolling more than any other part of that piece of material.

31. Material Requiring Treatment.—In case of boiler tubes, shafts, or other material, which occasionally require one or more retreatments, the objects must have attached sufficient material to enable the cutting of test pieces after each treatment. The number of submissions of material for inspection after treatment will be limited to three; the inspector being given the option of permitting a fourth submission in his discretion.

CHEMICAL ANALYSES.

32. Government's Analysis.—Drillings for analysis must be fine, clean, dry, and free from mill scale and taken in the presence of the inspector. The inspector may take them from the ingot, bloom, or billet; from any test piece, or from any part of the material, provided, in this case, that by so doing the material will not be rendered unfit for use. Unless otherwise directed, the chemist will make determinations of those elements only which are limited by the specifications.

33. Manufacturers' Analysis.—Before the inspector finally accepts material the manufacturers shall furnish him with a copy of their analysis of the same, and, when required by the inspector, the same shall be certified to.

MATERIAL EXEMPT FROM TESTS.

34. Class C Material.—This includes class C engine forgings, class C steel castings, and class C boiler plates, shapes, bolts, rivets, studs and nuts. This material will not be subject to tests or analysis unless there is reason to doubt that it is of a suitable quality.

The above-named material will be subject to surface inspection and to inspection as to dimensions. This inspection will be made at the manufacturers' works, or at the contractors' warehouse, or at the point of delivery.

The object of this class of material is to permit of the purchase of commercial articles with the least expense and delay for inspection, and for purposes where high-grade material is not needed.

ADDITIONAL TESTS.

35. By Bureau's Orders.—Tests may be prescribed by the Bureau of Steam Engineering for the inspection of material for which tests are not specified herein.

36. By Inspector's Decision.—The inspector may make, from time to time, such additional tests as he may deem necessary to determine the uniformity of the material.

37. Because of Insufficient Hammering or Other Treatment.—Whenever from the form of the ingot or other cause the inspector considers that certain parts of a forging have not received sufficient hammering or other treatment he will make additional tests of those parts.

38. Class C Material.—If doubt should arise as to the quality of class C

material not inspected at place of manufacture, the inspector at the point of delivery shall have the right to test it at the expense of the contractor, the material to be in accordance with the specifications for its particular class.

39. When Heat Number is Doubtful.—In case there is doubt as to the identity of the heat from which a plate, forging, or other object is made, such plate, forging, or other object shall be tested singly or in lots, and must conform to the chemical and physical requirements laid down for its class.

TENSILE TESTS AND TEST PIECES.

40. Interpretation.—The tensile strength herein specified means the ultimate tensile strength per square inch of original cross section. The elastic limit may be measured by the drop of the beam or the halt of the gauge of the testing machine. The elongation is that obtained after fracture. In the case of test pieces of rectangular section the reduction of area is to be measured by the product of the average width and thickness of the reduced area and not the minimum width and thickness.

41. Pulling Speed.—Each tensile-test piece shall be subjected to a direct tensile stress until it breaks, in a machine of standard manufacture, running at a pulling speed of not less than 1 inch and not more than 5 inches per minute for 8-inch test pieces, and not less than $\frac{1}{2}$ inch and not more than 3 inches per minute for 2-inch pieces.

42. Dimensions of Test Pieces.—Tensile-test pieces shall be of the types shown in the following figures:

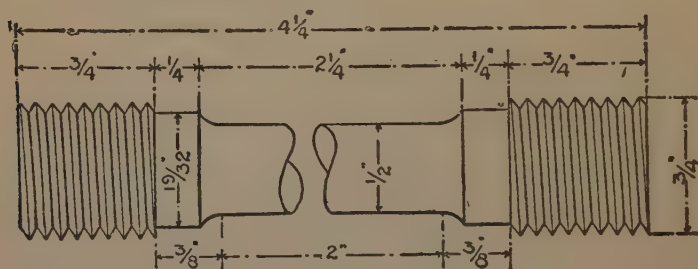


FIGURE 1. For round test bars.

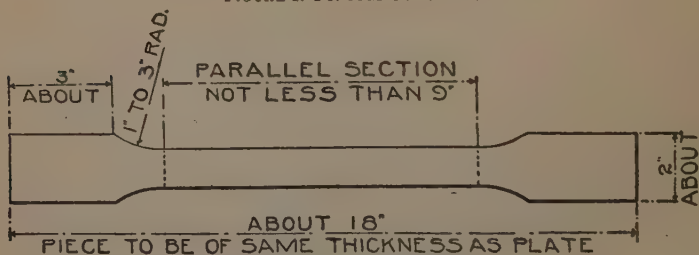


FIGURE 2. For flat test bars.

43. **Standard Length Between Measuring Points.**—Test pieces from blooms, large rolled bars exceeding 2 inches in diameter, forgings, and castings are to have a length between measuring points of 2 inches, as shown in figure 1 of article 42. Other test pieces are to have a length between measuring points of 8 inches, as shown in figure 2 of article 42, except as otherwise directed in these specifications.

44. **Allowance of Variation in Area of Test Pieces.**—A variation of 5 per cent above or below in area is allowed.

45. **Full-size Bars.**—Full-size bars and rods within the capacity of the testing machine may be used as tensile-test pieces, and in this case the bending tests may also be taken from the full-size bars and rods.

46. **Plates, Minimum Thickness.**—When heats are rolled into plates of varying thickness the test pieces shall be taken from plates of not less than 0.3 inch thick.

47. **Plates and Steam Pipes, Standard Size for Test Pieces.**—The width of tensile-test pieces from plates and steam pipes over 5/16 inch in thickness will be 1½ inches, the thickness the same as the plate or steam pipe, and the length between measuring points 8 inches; under 5/16 inch the width will be not over 2 inches, the thickness the same as the plate or steam pipe, and the length between measuring points the same as prescribed in articles 88 and 141, respectively.

BENDING TESTS.

48. **Rounding of Edges of Test Pieces.**—In the case of bending-test pieces of rectangular section the edges may be rounded off to a radius equal to one-fourth of the thickness.

49. **Standard.**—Bending-test pieces for blooms, large rolled bars (exceeding 2 inches in diameter), forgings, and castings shall be 1 inch wide by ½ inch thick. For plates the width shall be not less than 1½ inches and the thickness that of the plate. The bending may be done either by pressure or by blows.

ENGINE FORGINGS.

50. The physical and chemical characteristics are to be in accordance with the following table:

Class.	Material.	Treatment.	Minimum tensile strength (pounds per square inch).	Minimum elastic limit (pounds per square inch).	Minimum elongation (per cent in 2 inches).	Maximum percentage of—		Without showing cracks or flaws must cold bend about an inner diameter of—
						P.	S.	
H. G...	Open - hearth nickel steel.	Annealed and oil tempered.	95,000	65,000	21	.05	.05	1 inch through 180°.
A.....	Open - hearth, either nickel or carbon steel.	Annealed. Oil - tempering optional.	80,000	50,000	25	.05	.05	Do.
B.....	Open - hearth carbon steel.	Annealed.....	60,000	30,000	30	.05	.05	½ inch through 180°.
C.....	Open - hearth or Bessemer steel.	Do.....	52,000		28			1 inch through 180°.

NOTE.—Class C forgings will not be tested unless there is reason to doubt that they are of a quality suitable for the purpose for which intended. Tests, if required, shall be made at the expense of the contractor, and may be made at the point of delivery.

51. Treatment.—All forgings shall be annealed as a final process, unless otherwise directed. All tempered forgings, if forged solid, and if more than 5 inches in diameter in any part of their lengths, not including collars, palms, or flanges, shall be bored through axially before tempering, and the bore shall be of sufficient size to enable the manufacturer to get the requisite tempering effect. Forgings, such as crank shafts, thrust shafts, etc., may, previous to tempering, be machined in a manner best calculated to insure that the tempering effect reaches the desired portions. In this case the inspector will decide upon the location of the test pieces if they cannot be taken in the manner hereinafter described.

52. Surface and Other Defects.—All forgings shall be free from slag, cracks, blowholes, hard spots, sand, foreign substances, and all other defects affecting their value.

53. Kind of Ingot.—The tests herein laid down are adapted to exhibit the qualities of forgings made from the ordinary square, cylindrical, or polygonal ingots cast on end. If the contractor intends to use any other form he shall advise the Bureau of Steam Engineering beforehand in order that the location of the test bars may be arranged accordingly. If ingots are cast in any unusual manner, the amount of the discard from them will be determined by the Bureau of Steam Engineering with a view to leaving the portion to be used at least as good as the metal of an ingot cast in the ordinary way, from which a discard of 30 per cent from the top and 5 per cent from the bottom has been made, or if the ingot is bottom cast, a discard of 20 per cent from the top and 5 per cent from the bottom. If the inspector finds it impracticable or inadvisable to enforce the requirements with regard to the amount of discard from ingots, he shall take a sufficient number of tests from each end of every large forging to satisfy himself that the material is of uniform quality from end to end.

54. Uniformity Tests.—See article 36.

55. Test Pieces from Rotor, Line, Thrust, and Propeller Shafts.—From each length of rough-forged shaft and from the end which was uppermost in the ingot, one tensile-test piece shall be taken at a distance from the center equal to the radius of the finished shaft, and one tensile and one bending test piece shall be taken at half that distance from the center. From the other end of the same length of shaft one tensile-test piece shall be taken at a distance from the center equal to half the radius of the finished shaft. If the shaft is 10 or more inches in diameter, three tensile-test pieces shall be taken from the upper end of the shaft and two tensile-test pieces from the other end, the bending test piece being taken as in the case of the smaller shafts. At the option of the manufacturer the test pieces may be taken either parallel or at right angles to the axis of the shaft. In the case of hollow shafting (either forged or bored) the inside pieces shall be taken within the finished section prolonged, but as near as practicable to one-half the finished radius from

the center. If the couplings are forged on the shaft, the test pieces shall be taken from a prolongation of the shaft which shall have received the same reduction as the shaft at its greatest diameter.

56. Test Pieces from Crank Shafts.—Test pieces from such shafts shall be taken in the same manner and in the same number as described for rotor, line, thrust and propeller shafts, except that two additional test pieces shall be taken from each crank, one from the surface of the metal slotted out and one at a distance of one-half the finished radius of the shaft from the plane passing through the axes of the shaft and crank pin and both taken in a plane perpendicular to that last mentioned and passing through the axis of the ingot. In the case of crank shafts having more than one throw in one forging, these test pieces may be taken from one crank only. If desired, the piece between the arms of the crank may be slotted out before annealing, in which case the two test pieces shall be taken from spare material left on the crank pin on the side toward the axis of the shaft, one piece in a plane at one-half the finished radius of the shaft from the plane containing the axes of the shaft and pin; the other piece at or near the surface of the metal, but not near the position of the first piece. In the case of test pieces taken from crank webs, it will be satisfactory if the elongation falls 2 per cent below the elongation specified for the crank shaft.

57. Test Pieces from Piston Rods, Connecting Rods, Eccentric Rods, Valve Stems, Columns, Tie-rods, Wrist Pins, Crossheads, Valve Links, Guides, Forged Bolts and Nuts, Feather, Keys, Collars, Sleeves, Couplings, and Caps.—One longitudinal tensile-test piece shall be taken from the prolongation of one end of the heads or ends of the rough-forged rods, stems, etc., and one longitudinal cold-bending test piece shall be taken from the prolongation of the other end. If, however, the single finished forging weighs less than 100 pounds, the forgings may be tested in lots of 1000 pounds or less, the pieces in each lot being of one kind only, made from the same heat and annealed in the same furnace at the same time. In this case the inspector will select at random two longitudinal tensile and two longitudinal cold-bending test pieces to represent the lot, in accordance with instructions under "General test requirements." When the manufacturer so desires, extra forgings may be made in order to provide for test pieces, which will be selected by the inspector at random from the lot.

58. Test Pieces from Reverse Shafts.—If the shaft is 5 inches or less in diameter. One longitudinal tensile-test piece shall be taken from one end and one longitudinal cold-bending test piece shall be taken from the other end. If the shaft is over 5 inches in diameter, one tensile and one cold-bending test piece shall be taken from the end which was uppermost in the ingot, and one tensile-test piece from the other end.

59. Test Pieces from Forged Pistons, or Other Forgings of Irregular Shape, not Mentioned in the Foregoing Paragraphs.—The inspector shall

select one or more tensile and cold-bending test pieces, according to the size and shape of the forging. These pieces may be taken either from the forging or a prolongation of the same provided for the purpose.

60. Test Pieces from Miscellaneous Bars.—The inspector shall select four tensile and two cold-bending test pieces from each heat as rolled or forged from material annealed in the same furnace at the same time. In each case where material is to be reformed the contractor shall state the fact on his order. Hot-bed annealing will be satisfactory for hot-rolled material.

INGOTS, SLABS, BLOOMS, AND BILLETS.

(Rounds shall be classed as billets if they are to be reformed. The line between blooms and billets to be drawn at size of 5 inches square.)

61. Ingots, slabs, blooms, and billets made by steel manufacturers and to be forged or rolled into finished objects by them will not require inspection or tests; the tests and inspection will be made of the finished objects.

62. Ingots made by steel manufacturers and to be forged into finished objects by establishments other than those manufacturing the ingot will be subjected to chemical test and surface inspection only at place of manufacture. All required physical tests will be made from the finished objects, as previously directed.

63. Slabs, blooms, and billets from which small objects are to be machined without heating shall be tested by heats, four longitudinal tensile and four longitudinal cold-bending test pieces being selected, each from a different object; but if less than 10 pieces are made from one heat, then two tensile and two cold-bending test pieces shall be selected; but if there is but one slab, bloom, or billet from a heat, one longitudinal tensile and one longitudinal cold-bending test piece will suffice, either or both test pieces to be taken from upper or lower end at discretion of the inspector. All of these test pieces shall be selected in accordance with the "General Instructions" concerning tests and test pieces, and the finished objects will not require any further tests. These slabs, blooms, and billets may be tempered and annealed, or only annealed, at the discretion of the manufacturer, to get the physical requirements, and these requirements shall be the same as for the class of forgings for which the objects are intended. See article 25.

64. Slabs, blooms, and billets to be forged into finished objects by establishments other than those manufacturing them shall be tested by heats, four longitudinal tensile and two longitudinal cold-bending test pieces being selected, each from a different object; but if less than 10 pieces are made from a heat, then two tensile and two bending test pieces will suffice. And if there is but one slab, bloom, or billet from a heat, test pieces will be taken as in similar cases noted in article 63.

The requirements for class H. G., class A and class B slabs, blooms,

and billets which are to be reformed shall be the same as for engine forgings of the same class, except that an elongation of 24 per cent in 2 inches will suffice.

65. The contractors shall state on their orders whether or not the ingots, slabs, blooms, or billets are to be again worked hot. If they are to be worked hot, the pieces, before shipment, must be plainly marked "For reforming."

DROP FORGINGS.

66. Drop forgings of parts subject to pressure or to strains must meet the physical and chemical requirements of class B forgings.

67. Drop forgings of parts not subject to pressure to have surface inspection only.

STEEL CASTINGS.

68. The physical and chemical characteristics of steel castings are to be in accordance with the following table:

Class.	Material.	Treatment.	Minimum tensile strength (pounds per square inch).	Minimum elastic limit (pounds per square inch).	Minimum elongation (per cent in 2 inches).	Minimum reduction of area (per cent).	Maximum percentage of P.	Cold bend about an inner diameter of 1 inch.
A.....	Any approved process.	Annealed.	80,000	35,000	15	20	0.36	Through 90°.
B.....	do.....	do.....	65,000	30,000	20	25	.06	Through 120°.
C.....	do.....	do.....						

NOTE.—Class C castings will ordinarily be subject to surface inspection only, at the point of delivery. They will not be tested unless there are reasons to doubt that they are of a quality suitable for the purpose for which they are intended. Tests, if required, shall be made at the expense of the contractor, and may be made at the point of delivery. The inspector will select a sufficient number of the castings and have them crushed, bent, or broken, and note their behavior and the appearance of the fracture.

69. Kind of Material.—Steel for castings may be made by the crucible, open-hearth, Bessemer, Tropenas, or any other process approved by the Bureau of Steam Engineering. The material must be of uniform quality throughout. In case of doubt on this point, the inspector will have drillings for complete analysis taken from the surface and from the center of the thickest parts. In all cases analyses will be made for phosphorus, and no casting shall contain more than six one-hundredths of 1 per cent of that element.

70. Treatment.—All castings shall be annealed in accordance with articles 24 and 25 of these specifications, unless otherwise directed. If castings are subjected to any special treatment, the inspector will make

such additional tests as he may deem necessary to see that the treatment has left the material of uniform character throughout. In case the results obtained by the first tests do not conform to the specifications, the manufacturer may re-treat the castings and submit additional pieces for test.

71. Surface and Other Defects.—All castings shall be thoroughly cleaned before they are presented for inspection. They shall be sound, free from brittleness, injurious roughness, sponginess, pitting, porosity, shrinkage and other cracks, cavities, foreign substances, and all other injurious defects. They shall be tested with a hand hammer. Particular search shall be made at the points where the heads or risers join the castings, as unsoundness at such points is likely to extend into the castings.

72. Test Pieces.—Sound test pieces shall be taken in sufficient number to thoroughly exhibit the character of the metal in the entire piece from each of the following castings, viz: Shaft struts or brackets, main cylinder or valve chest liners, main pistons and followers; eccentric, reversing, and rock-shaft arms; crossheads, bedplates, columns of main engines and main air pumps, shaft couplings, and all large castings weighing over 200 pounds. All other castings may be tested by lots, as follows: A lot shall consist of all castings from the same heat, annealed in the same furnace charge. From each lot two or more tensile and one or more bending test pieces shall be taken, and the lot passed or rejected on the results shown by the tests. The test pieces may, at the discretion of the inspector, be cut either from coupons to be molded and attached to some portion of the casting, or else from sinking heads, in cases where such heads of sufficient size are employed. Coupons are to be attached so as not to interfere with the successful making of the casting, but at the same time to show the quality of the material. In the case of castings tested by lots, and if the manufacturer so desires, the test pieces may be taken from the body of a casting selected by the inspector from the lot. In no case shall the coupons be detached from their castings before they are stamped by the inspector. In special cases where the attachment of a coupon would be liable to injure the casting, the coupon may be cast with small runners to the gate, or separately in the same flask, so that it may be easily identified when the casting is knocked out of the sand, which must be done in the presence of the inspector. In all such cases the sanction of the inspector must first be obtained.

IRON CASTINGS.

(For cylinder casings, valve chests, and other important parts.)

73. The inspection must conform to the General Instructions for the Inspection of Material.

74. All engine castings must be in accordance with the machinery specifications, and must finish to blue-print size.

75. The grade and quality of the metal will be specified on the order.

76. The castings must be smooth, free from blowholes, porous places, shrinkage, or other cracks or defects, and must be well cleaned.

77. The minimum tensile strength of the material shall be 20,000 pounds per square inch, the length of the test piece being not less than 2 inches. The transverse breaking load for a bar 1 inch square, loaded at the middle and resting on supports 1 foot apart, shall be not less than 2200 pounds for any casting, nor more than 2800 pounds for main engine cylinder casings and valve chests.

78. The scale shall be removed from the unfinished parts of the insides of all cylinders, cylinder covers, and valve-chest covers, and from the unfinished parts of all cylinder and valve-chest liners, and from ports and passages of cylinders and valve chests, either by pickling or other approved process as may be required by the machinery specifications.

79. For cylinder liners and valve-chest liners the following additional requirement must be met: The transverse breaking load for a bar 1 inch square, loaded at the middle and resting on supports 1 foot apart, shall be not less than 2500 pounds.

80. Great care must be taken to determine that the machinery specifications for hardness of cylinders, liners, and valve-chest liners are complied with. A test piece from the casting should be machined in order to show the degree of hardness.

IRON CASTINGS (CLASS C).

81. Iron castings (class C) for minor parts, such as furnace fittings, etc., will be subject to inspection at the point of delivery only to see if they are in all respects suitable for the purposes for which they are intended.

MALLEABLE CASTINGS.

82. Malleable-iron castings for which physical requirements are specified may be made by either the open-hearth or the air-furnace process.

83. Sulphur must not exceed 0.06 per cent, and phosphorus must not exceed 0.225 per cent.

84. The transverse breaking load for a bar 1 inch square, loaded at the middle and resting on supports 1 foot apart, shall be not less than 3000 pounds, deflection being at least $\frac{1}{2}$ inch.

85. The minimum tensile strength of the material must be 40,000 pounds per square inch, and the elongation at least $2\frac{1}{2}$ per cent in 2 inches.

86. Castings must be true to pattern, free from blemishes, scale, and shrinkage cracks.

87. Malleable castings must be neither "over" nor "under" annealed. They must have received their full heat in the oven at least 60 hours after reaching that temperature, and shall not be dumped until they are at least "black hot."

BOILER PLATE.

88. The physical and chemical characteristics of steel boiler plate are to be in accordance with the following table:

Class.	Material.	Tensile strength (pounds per square inch).	Minimum elastic limit (pounds per square inch).	Elongation (per cent in 8 inches).	Maximum percentage of—		Without showing cracks or flaws must cold bend about an inner diameter—
					P.	S.	
A	Open - hearth steel.	65,000 to 73,000	$\frac{1}{2}$ T. S...	22	0.04	0.035	Equal to thickness of plate and through 180° for plates 1 inch in thickness and under and equal to one and one-half times the thickness through 180° for plates over 1 inch in thickness.
B	do	58,000 to 65,000	$\frac{1}{2}$ T. S...	25	.04	.035	Flat back through 180° for plates under 1 inch in thickness and equal to thickness of plate through 180° for plates 1 inch and over in thickness.
C	Open hearth or Bessemer steel.	To be in accordance with specifications for flange and boiler steel adopted by the Association of American Steel Manufacturers, revised 1903.					

NOTE.—When the finished plate is $\frac{1}{2}$ inch or less the elongation shall be measured on an original length of sixteen times the thickness of the plate tested.

When plates are ordered to gauge, United States standard gauge will be used.

89. Surface and Other Defects.—Plates must be free from all slag, foreign substances, brittleness, laminations, hard spots, brick or scale marks, scabs, snakes, or other injurious defects.

90. Test Pieces.—One longitudinal tensile-test piece and one bending-test piece (transverse for class A and class B material and longitudinal for class C material) must be cut from each plate as rolled at such points as may be designated by the inspector. The cold-bending test pieces may have their corners rounded to a curve the radius of which is equal to one-fourth the thickness of the plate. In case a plate is to be reheated after leaving the works of the manufacturer, the order to the manufacturer shall state this fact and also it shall specify the plate of sufficient size to provide for additional test pieces to be cut from the plate after it has been reheated.

91. Shearing.—Boiler plates shall not be sheared closer to finished dimensions than once the thickness of the plate along each end and one-half the thickness of the plate along each side. This allowance shall be made by the contractor on his order, and the manufacturer shall shear to the ordered dimensions.

92. Gauge.—A tolerance of 0.01 inch below the ordered gauge will be

permitted for plates up to and including 100 inches in width, and for plates over 100 inches in width a tolerance of 0.015 inch will be allowed, measured in each case at the thinnest point.

93. Weight.—For all plates ordered to gauge there will be permitted an average excess of weight over the calculated weight equal in amount to that specified in the following table:

ALLOWANCES FOR OVERWEIGHT FOR PLATES WHEN ORDERED TO GAUGE.

Thickness of plate in inches.	Width of plate in inches.					
	Up to 50.	50 to 70.	Over 70.	Up to 75.	75 to 100.	100 to 115.
	Percentage allowed.					
Under $\frac{5}{16}$	10	15	20			
$\frac{5}{16}$ up to $\frac{3}{8}$	$8\frac{1}{2}$	$12\frac{1}{2}$	17			
$\frac{3}{8}$ up to $\frac{1}{2}$	7	10	15			
$\frac{1}{2}$				10	14	18
$\frac{5}{8}$				8	12	16
$\frac{3}{4}$				7	10	13
$\frac{7}{8}$				6	8	10
Over $\frac{1}{2}$				5	7	9
.....				$4\frac{1}{2}$	$6\frac{1}{2}$	$8\frac{1}{2}$
.....				4	6	8
Over $\frac{3}{8}$				$3\frac{1}{2}$	5	$6\frac{1}{2}$

The weight of 1 cubic inch of rolled steel is assumed to be 0.2833 pound.

BOILER PLATES AND SHAPES (CLASS C).

94. Under this head will be included all material of that kind which is not essential to the structural strength of boilers and engines, such as:

Air ducts.	Fireroom air screens.
Ash dumps.	Furnace doors.
Ash pans.	Ladders.
Ash-pit doors.	Oil tanks.
Blower casings and fans.	Platforms and gratings.
Boiler casing, including that for water-tube boilers.	Smoke pipes and covers.
Circulating plates for boilers.	Tallow and other tanks.
Coal and ash buckets.	Uptakes and uptake doors.
	Feed and filter tanks.

NOTE.—Neither chemical nor physical tests will be required of these items.

95. Grate bars are to be free from injurious defects and to conform to the dimensions ordered; maximum variation in thickness, plus or minus, 0.009 inch.

BOILER BRACING.

96. Kind of Material.—Steel for boiler bracing shall be made by the open-hearth process and shall not show more than four one-hundredths of 1 per cent of phosphorus or more than four one-hundredths of 1 per cent of sulphur, and shall be of the best composition in other respects. A variation in cross section or in weight of more than 5 per cent over that specified will be sufficient cause for rejection.

One ton of material for boiler bracing, from the same heat, shall constitute a lot from which two tensile and one cold-bending test specimens shall be taken, each from a different object.

97. Treatment.—All forged material for boiler bracing shall be annealed as a final process.

98. Tensile Tests of Class A Bracing.—Test specimens must show a tensile strength of at least 75,000 pounds per square inch, with an elastic limit of at least 40,000 pounds per square inch, and an elongation of at least 23 per cent in 8 inches, or 26 per cent in 2 inches in case 8-inch specimens cannot be secured.

99. Tensile Tests of Class B Bracing.—Test specimens must show a tensile strength of at least 60,000 pounds per square inch, with an elastic limit of at least 32,000 pounds per square inch, and an elongation of at least 26 per cent in 8 inches, or 30 per cent in 2 inches in case 8-inch specimens cannot be secured.

100. Cold-bending Tests.—One test piece cut from each lot of Class A shapes for bracing must stand bending cold through an angle of 180° to an inner diameter equal to twice the thickness of the piece tested without showing a fracture on outside of bent portion.

One bar $\frac{1}{2}$ inch thick, cut from each lot of class A forgings for bracing, must stand cold bending through an angle of 180° to an inner diameter of 1 inch without showing a fracture on outside of bent portion.

One test piece cut from each lot of class B shapes for bracing must stand bending cold through an angle of 180° to an inner diameter equal to once the thickness of the piece tested without showing a fracture on outside of bent portion.

One bar $\frac{1}{2}$ inch thick, cut from each lot of class B forgings for bracing, must stand cold bending through an angle of 180° to an inner diameter of $\frac{1}{2}$ inch without showing a fracture on outside of bent portion.

101. Opening and Closing Tests.—Angles, tee bars, and other shapes for boiler bracing are to be subjected to the following additional tests: A piece cut from one bar in twenty shall be opened out flat while cold without showing cracks or flaws; a piece cut from another bar in the

same lot shall be closed down on itself until the two sides touch without showing cracks or flaws.

102. **Inspection for Surface and Other Defects.**—All boiler bracing must be true to form, free from seams, hard spots, brittleness, injurious sand or scale marks, and injurious defects generally.

FLOOR PLATES.

103. **Floor Plates.**—Floor plates to be made from steel plates. They must be free from surface defects and conform to dimensions ordered. Unless ordered with planed edges, plates will be shop sheared and a variation of $\frac{1}{8}$ inch in dimensions will be allowed.

(a) They shall be of ribbed or diamond pattern, as ordered.

(b) Ribs and diamonds shall be symmetrical, well defined, with sharp edges and flat tops, and the axes of patterns shall be parallel with the longest dimension. Length of diamond not to exceed $3\frac{1}{2}$ inches at the bottom. In ribbed plates, the ribs shall cover approximately half the surface.

(c) The underside of the plates shall be flat and reasonably free from marks of rolls.

DIMENSIONS AND WEIGHTS OF PLATES.

Thickness at bottom of pattern; minimum.	Height of rib or diamond.	Weight for diamond pattern; maximum.	Weight of ribbed pattern; maximum.
<i>Inch.</i>	<i>Inch.</i>	<i>Pounds.</i>	<i>Pounds.</i>
$\frac{1}{8}$	$\frac{3}{8}$	8	9
$\frac{3}{16}$	$\frac{3}{8}$	8 $\frac{1}{2}$	9 $\frac{1}{2}$
$\frac{1}{4}$	$\frac{1}{2}$	11 $\frac{1}{2}$	12 $\frac{1}{2}$
$\frac{5}{16}$	$\frac{1}{2}$	14	15
$\frac{3}{8}$	$\frac{1}{2}$	16 $\frac{1}{2}$	17 $\frac{1}{2}$
$\frac{7}{16}$	$\frac{1}{2}$	19 $\frac{1}{2}$	20 $\frac{1}{2}$
$\frac{1}{2}$	$\frac{1}{2}$	21 $\frac{1}{2}$	22 $\frac{1}{2}$

(d) Plates exceeding weight in table by not more than 5 per cent may be accepted, but excess weight will not be paid for. A minus variation in height of rib or diamond will be allowed as follows:

	<i>Inch.</i>
Plates up to 36 inches wide.....	0.01
Plates 36 inches wide and over.....	.03

No upper limit is placed on the height.

RODS FOR BOLTS, STUDS, AND RIVETS.

104. The physical and chemical characteristics of rods for bolts, studs, and rivets are to be in accordance with the following table:

Class.	Material.	Minimum tensile strength (pounds per square inch).	Minimum elastic limit (pounds per square inch).	Minimum elongation ¹ (per cent in 8 inches).	Maximum amount of—		Bends. ²
					P.	S.	
A.....	Open - hearth nickel or carbon steel.	75,000	40,000	23	0.04	0.035	Cold bend 180° about an inner diameter equal to one-half the thickness of the test piece for diameters up to and including 1 inch, and equal the thickness for diameters over 1 inch; quench bend 180° about an inner diameter equal to the thickness of the test piece for diameters up to and including 1 inch and equal to 1½ times the thickness for diameters over 1 inch.
B.....	Open - hearth carbon steel.	58,000	30,000	28	.04	.035	Cold bend flat back through 180°; quench bend 180° through an inner diameter equal to one-half the thickness of the test piece for diameters up to and including 1 inch and equal to the thickness for diameters over 1 inch.
C.....	Open - hearth steel.	To be in accordance with specifications for rivet steel adopted by the Association of American Steel Manufacturers, revised 1903.					

¹ Elongation for rounds $\frac{1}{4}$ inch and less in diameter shall be measured in an original length equal to 16 times the diameter of the test piece; for material over $\frac{1}{4}$ inch up to and including 1 inch in diameter, the elongation shall be measured in a length of 8 inches; and for material over 1 inch in diameter up to and including 2 inches in diameter, the required percentage of elongation, measured in a length of 8 inches, shall be reduced by one for each increase in diameter of $\frac{1}{4}$ inch or a fraction thereof above 1 inch.

Requirement for minimum elastic limit will not apply to material for rivets.

² Quench test pieces to be heated to a dark cherry red, as seen in daylight, and plunged into fresh clean water of 80° to 90° F.

If the contractor desires, and so states on his order, the bureau will direct that the inspection of the rods be made at the place of manufacture of the bolts, studs, or rivets instead of at the place where the rods are rolled.

105. Surface and Other Defects.—The rods must be true to form, free from seams, hard spots, brittleness, injurious sand or scale marks, and injurious defects generally.

106. Tensile Strength.—One tensile-test piece shall be taken from each ton or fraction thereof of rods rolled from the same heat. If, however, the rods in one heat are not of the same diameter, then the inspector will take such additional test pieces as he may consider necessary according to the number of different sizes of rods in the heat. When practicable,

but one piece will be cut from each rod selected for the test. Should any test piece be found too large in diameter for the testing machine, the piece may be prepared for test in the manner prescribed for forgings.

107. Bending Tests.—If the total weight of rods rolled from the same heat amounts to 6 tons or more, four cold-bending test pieces and four quench-bending test pieces must be taken; but if the weight is less than 6 tons, one-half of that number of test pieces will suffice.

108. Upsetting Tests.—From each heat of rounds as rolled there shall be cut six test specimens about $1\frac{1}{2}$ inches long, which shall stand hammering down cold, longitudinally, to one-half their original length without showing seams or other defects which would tend to produce imperfections in the finished product.

FINISHED BOLTS, STUDS, AND RIVETS (CLASSES A AND B).

109. After the rods to be made up into bolts, studs, or rivets have been tested as previously described, the finished articles shall be tested by lots of 500 pounds or fraction thereof, one piece being taken to represent the lot. The failure of 10 per cent of the lots of 500 pounds to stand the specified tests in a satisfactory manner will render the whole of any delivery liable to rejection.

110. Bolts and Studs.—When the bolts or studs are of sufficient length in the plain part to admit of being bent cold, they must stand bending double to a curve of which the inner radius is equal to the radius of the bolt or stud, without fracture.

111. When bolts or studs are not of sufficient length in the plain part to admit of being bent cold, the threaded part must stand bending cold without fracture as follows:

	Degrees.
If of $\frac{1}{2}$ inch diameter or less.....	35
If above $\frac{1}{2}$ inch diameter and under 1 inch.....	30
If 1 inch diameter or over.....	25

112. Where the bending tests cannot be applied, the two following hammer tests must be substituted:

(a) The test piece to stand flattening out cold to a thickness equal to one-half its original diameter without showing cracks.

(b) The test piece, while heated to a cherry-red heat in daylight, to stand flattening out to a thickness equal to one-third its original diameter without showing cracks.

113. Surface Inspection.—(1) All bolts and studs must be free from surface defects.

(2) All bolts must be headed hot, and the heads made in accordance with the United States standard proportions unless otherwise specified. The head must be concentric with the body of the bolt.

(3) The threads must be of the United States standard unless otherwise specified, and must be clean and sharp. The threads of classes A

and B bolts may be either chased or cut with a die, but the threads of body-bound bolts must be chased and must extend far enough down to insure that the distance between the head and the nut does not exceed one and one-half threads when the nut is screwed home. The plain part of body-bound bolts must be turned in a lathe to fit accurately in the bolt hole.

114. Rivets.—Samples from each lot must stand the following tests without fracture, test (a) being applied to one lot, and (b) to a second, etc.:

(a) Bend double cold to a curve of which the inner diameter is equal to the diameter of the rivet.

(b) Bend double hot through an angle of 180° flat back.

(c) The head to be flattened when hot without cracking at the edges until its diameter is two and one-half times the diameter of the shank.

(d) The shanks of sample rivets to be nicked on one side and bent cold to show the quality of the material.

115. Surface Inspection.—Rivets must be true to form, concentric and free from injurious scale, fins, seams, and all other injurious defects. If the material is found to be very uniform and none of the tests made of a series of lots fail, the inspector may discontinue the tests after he has made enough to satisfy himself that the whole of the material on the order is satisfactory.

NOTE.—In measuring the diameter of rivets the inspector will allow for the trade custom of making rivets with an actual diameter slightly (about 1/16 of an inch) less than the nominal diameter.

FINISHED BOLTS, STUDS, AND RIVETS (CLASS C).

116. Class C bolts, studs, and rivets may be used only with the plates and shapes enumerated in article 94.

STEEL AND IRON NUTS.

(To be used with class A and B bolts and studs.)

117. One tensile and one bending test bar from each lot of 1000 pounds of material or less from which nuts are to be made must be selected by the inspector for test.

118. The material (whether steel or iron) shall show a tensile strength of at least 48,000 pounds per square inch and an elongation of at least 26 per cent in 8 inches. A bar $\frac{1}{2}$ inch square or $\frac{1}{2}$ inch in diameter must bend back cold through an angle of 180° without showing signs of fracture.

119. The nuts must be free from surface defects and the threads clean, sharp, and well fitting.

120. The dimensions of threads must be in conformity with the United States standard unless otherwise specified.

STANDARD DIMENSIONS OF BOLTS AND NUTS FOR THE UNITED STATES NAVY.

Diameter.		Area.	Threads.	Long diameter.		Short diameter.	Depth.		Diameter of holes in blank nuts.
Nom.	Eff.	Eff.	No.	Hex.	Sq.	W.	Head.	Nut.	
$\frac{1}{8}$	0.185	0.026	20	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{7}{16}$
$\frac{1}{16}$	.240	.045	18	$\frac{1}{16}$	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{4}$
$\frac{3}{16}$	.294	.067	16	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{3}{16}$	$\frac{1}{4}$
$\frac{7}{16}$	.345	.093	14	$\frac{3}{16}$	$1\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{7}{16}$	$\frac{1}{4}$
$\frac{1}{2}$	.400	.125	13	1	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{3}{4}$
$\frac{9}{16}$	.454	.162	12	$1\frac{1}{8}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{9}{16}$	$\frac{3}{4}$
$\frac{5}{8}$	.507	.202	11	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{3}{4}$	.620	.302	10	$1\frac{7}{8}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$
$\frac{7}{8}$	.731	.419	9	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{7}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{2}$
1	.837	.550	8	$1\frac{7}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$	$\frac{1}{2}$	1	$\frac{5}{8}$
$1\frac{1}{8}$	.940	.694	7	$2\frac{3}{8}$	$2\frac{1}{8}$	$1\frac{3}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$
$1\frac{1}{4}$	1.065	.891	7	$2\frac{1}{2}$	$2\frac{3}{4}$	2	1	$1\frac{1}{4}$	$1\frac{1}{8}$
$1\frac{3}{8}$	1.160	1.057	6	$2\frac{1}{2}$	$3\frac{1}{4}$	$2\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{8}$
$1\frac{1}{2}$	1.234	1.294	6	$2\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{3}{8}$
$1\frac{5}{8}$	1.389	1.515	$5\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{3}{8}$
$1\frac{3}{4}$	1.491	1.746	5	$3\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{2}$
$1\frac{7}{8}$	1.616	2.051	5	$3\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{7}{8}$	$1\frac{5}{8}$
2	1.712	2.302	$4\frac{1}{2}$	$3\frac{1}{2}$	$4\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{5}{8}$	2	$1\frac{3}{4}$
$2\frac{1}{4}$	1.962	3.023	$4\frac{1}{2}$	$4\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{3}{4}$
$2\frac{1}{2}$	2.176	3.719	4	$4\frac{1}{2}$	$5\frac{1}{2}$	$3\frac{7}{8}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{8}$
$2\frac{3}{4}$	2.426	4.622	4	$4\frac{1}{2}$	6	$4\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{8}$

121. The nuts must be hot-pressed or cold-punched, the latter to be reamed before threading, the holes to be central and square with the faces. All nuts must fit on the bolts without shake.

122. Nuts to be used about machinery must fit so tight that it will be necessary to use a wrench to turn them. All other nuts must be at least thumb-tight.

123. For the purpose of test all nuts which fulfill the preceding requirements will be divided into lots of 500 pounds or less, and two nuts from each lot selected by the inspector for test as follows:

(a) One of the two must stand flattening out cold to a thickness equal to one-half its original thickness without showing cracks.

(b) The other must stand flattening out (when heated to a cherry red in daylight) to a thickness equal to one-third of its original thickness without showing cracks.

124. The failure to stand these tests will subject the lot represented by them to rejection. The failure of 10 per cent of the lots to pass the tests will render the whole order liable to rejection.

BOILER TUBES—GENERAL REQUIREMENTS.

125. Surface Inspection and Gauging.—Tubes must be free from all injurious surface defects, and in no part more than 6 per cent below the specified thickness. The defects to be particularly avoided in seamless cold-drawn tubes are tears, snakes, checks, slivers, injurious scratches, rings, laps, pits, and sinks; in hot-rolled tubes they are slivers, scratches, seams or laps, and scale pits; in lap-welded tubes they are defective welds, cracks, blisters, scale pits, and sand marks. If the tube ends are swaged down, swelled, upset, or reenforced, they will be subjected to a second inspection to see that no damage has been done by this process.

126. Weight.—A tolerance of 5 per cent over and 3 per cent under the calculated weight will be allowed for individual seamless cold-drawn and lap-welded tubes and 7 per cent over and 2 per cent under for seamless hot-rolled tubes, assuming that 1 cubic inch of iron weighs .280 pound and 1 cubic inch of steel .283 pound.

At the option of the inspector, tubes will be weighed in lots and the weight of each lot considered to be a multiple of the weight of each tube in that lot, the number of tubes in each lot being designated by the inspector.

If, however, the inspector or the manufacturer so desires, every tube of any size may be weighed separately, and accepted or rejected on its merits.

127. All seamless cold-drawn steel tubes must be annealed as a final process in retorts or in an approved furnace in which the flame does not strike the tubes. The amount of scale due to the annealing on the inside of the boiler tube shall be reduced as low as possible.

128. Tests.—(a) A test piece cut from each end of each tube must stand flattening under a press or hammer, immediately on being separated from the tube, without showing any defects on the outside.

(b) Of these flattened pieces 10 per cent or more, each cut from a different tube, must, on being opened for inspection, show no defects on the inside.

(c) Tubes will be divided into lots of not more than 100 (annealed in the same furnace at the same time for seamless cold-drawn tubes), and two tubes from each lot will be subjected to the following tests:

The ends of each tube to stand expanding cold to one and one-sixth times the original diameter and have a flange turned all around the end of the tube equal in width to one-eighth of the diameter of the tube without showing defects. If only one of the tubes so tested fails, that tube will be rejected, and other tubes will be selected, at the discretion of the inspector, from the same lot and subjected to test. These tubes must be found satisfactory in order that the lot may be passed.

(d) Each seamless tube will be subjected, in the presence of the inspector, to a hydrostatic pressure of 1500 pounds and each lap-welded tube to 1000 pounds per square inch, without showing signs of weakness or defects. The pressure will be applied by means of an accumulator or other method that will make the pressure steady during its application.

Failure to pass any of these tests may reject the entire lot of tubes.

129. The failure of 10 per cent of any lot of tubes to pass the required surface inspection or gauging will render that lot liable to rejection.

130. The manufacturers shall furnish for the bureau's records a quarterly statement giving a copy of their analyses of material used for naval boiler tubes.

131. Each tube must be stenciled with the name of the vessel for which it is intended (or the number of the government contract, bureau order, or requisition), the word "steel" or "iron," the maker's name, the inspector's initials, and, when necessary, the location mark.

132. After the inspection is completed and before shipment all tubes must be coated inside and outside with boiled linseed oil, or other preservative approved by the Bureau of Steam Engineering, to protect them from rust until such time as they are to be put into the boilers, when said coating will be removed.

ADDITIONAL TESTS FOR GENERATING TUBES FOR NICKLAUSSE BOILERS.

133. In addition to tests (a), (b), and (c), of article 128, a tensile test piece will be taken from one tube out of every lot of 100 tubes or less. These pieces must be machined to size about $1\frac{1}{2}$ inches wide and must show a tensile strength of at least 50,000 pounds per square inch and an elongation of at least 25 per cent measured over an original distance equal to 16 times the thickness of the test piece.

MATERIAL FOR BOILER TUBES.

134. (a) Seamless tubes must be made from solid disks or billets of perfectly homogeneous steel, by the process known as "cold drawing," and not by hot drawing with a cold finish, or they may be hot rolled.

(b) Lap-welded steel tubes must be made of steel of a perfectly homogeneous quality and of the grade which is found to insure the soundest welds.

(c) Iron tubes must be made of knobbled hammered charcoal iron; pig iron only (and no scrap) is to be used in the manufacture of the blooms.

STEAM AND WATER PIPES OF STEEL.

(Seamless and lap-welded.)

135. Material.—(a) The seamless pipes must be made from solid billets or disks of perfectly homogeneous steel, and cold drawn, hot rolled, or hot drawn, as specified in the order.

(b) The lap-welded steel pipes must be made of steel of a homogeneous quality and of the grade which is found to insure the soundest welds.

136. Surface Inspection.—The pipe must be free inside and outside from all surface defects which would materially weaken it or from starting points of corrosion. The defects to be especially avoided are the same as those mentioned under the head of defects in boiler tubes. The pipe must be perfectly round and vary not more than 1 per cent above or below the diameter specified.

137. The following tolerances will be allowed below the specified thickness:

Pipe 3 inches outside diameter and under, tolerance 6 per cent.

Pipe over 3 inches up to and including 5 inches outside diameter, tolerance 8 per cent.

Pipe over 5 inches up to and including $7\frac{1}{2}$ inches outside diameter, tolerance 10 per cent.

Pipe over $7\frac{1}{2}$ inches outside diameter, tolerance 12 per cent.

138. Weight.—The weight will be calculated from the specified dimensions and thickness, assuming that 1 cubic inch of steel weighs 0.283 pound. A tolerance of 3 per cent under and 7 per cent over weight will be allowed on all pipe except seamless pipe 4 inches or less in diameter, for which the allowance will be 3 per cent under and 5 per cent over weight.

139. Hydrostatic Test.—Every pipe must be subjected to an internal water pressure of 1000 pounds per square inch, without showing signs of bulges, cracks, or other defects.

140. Other Tests.—Pipe 4 inches outside diameter and less must be tested in the same manner as prescribed for steel boiler tubes.

141. Each pipe over 4 inches outside diameter must be tested as follows:

(a) A test piece must be cut circumferentially from one end of each pipe, straightened when hot, and machined to shape and to a width of $1\frac{1}{2}$ inches. For thicknesses up to and including $\frac{5}{16}$ inch, the length between the measuring points will be sixteen times the thickness. For thicknesses greater than $\frac{5}{16}$ inch, the length between measuring points will be 8 inches. In the case of welded pipe, the test piece must be cut so as to leave the weld in the middle. The test must show a minimum tensile strength of 48,000 pounds per square inch, and an elongation of at least 12 per cent in 8 inches or 15 per cent in 5 inches or less.

(b) A test piece of a width equal to three times the thickness of the

metal in the pipe, cut circumferentially from the end of each pipe (opposite to that from which the tensile piece was cut), so as to include the weld, if any, must stand bending cold at the weld to an inner diameter equal to thrice the thickness of the metal without showing cracks on the outside of the bend.

142. If the ends of the pipes are flanged, swaged down, swelled, upset, or reenforced, they will be inspected after this work is completed to see that no damage has been done by this process.

143. All cold-drawn pipes must be annealed in retorts or in an approved furnace in which the flame does not strike them.

144. Each pipe must be stenciled with the name of the vessel for which it is intended (or the number of the government contract, bureau order, or requisition), the maker's name, the inspector's initials, and when necessary, the location mark.

STEEL FLANGES FOR STEEL PIPE.

145. The flanges are to be forged from the solid (not welded) and are to be secured on the pipe as required by the order.

The steel of which flanges are made shall stand the following tests: The test piece to be taken from the ingot, billet, or plate from which the flange is forged, and to be reduced to shape with the least amount of hammering possible. One tensile-test piece and one bend shall be taken from each lot of 1000 pounds or less.

The test shall show that the metal has a tensile strength of at least 56,000 pounds per square inch, and an elongation of at least 30 per cent in 2 inches. The bending-test piece $1\frac{1}{2}$ inches wide must stand bending double (until the ends are parallel) without fracture (on the outside of the bend) to a curve of which the inner radius is one and one-half times the thickness of the test piece.

WROUGHT IRON.

(Used principally for parts which have to be welded.)

146. Wrought-iron bars and rods will be of best quality and manufacture and free from admixture of scrap or steel, and short pieces will not be used in piling. The finished material will be free from injurious defects.

147. The physical and chemical requirements will be as follows:

(a) Minimum tensile strength, 45,000 pounds per square inch.

(b) Minimum elastic limit, one-half tensile strength.

(c) Minimum elongation, 22 per cent. The elongation will be measured on an original length of 8 inches, except finished material below $\frac{3}{8}$ inch in diameter or thickness shall be measured on an original length of twelve times the diameter or thickness of section tested. Where a frac-

tional length results from the above, the length will be taken to the nearest inch.

(d) Minimum contraction of area, 40 per cent for rounds and 35 per cent for flats.

(e) *Tensile Test Piece*.—Material will be tested in sizes rolled when practicable, a sufficient number of tests to be taken to exhibit thoroughly the character of the material. When dimensions of bars or rods will not permit testing in sizes rolled, standard test piece will be used.

(f) *Nick Test*.—A bar nicked approximately 20 per cent of its thickness and bent back at this point through an angle of 180° must show a long, clean, silky fiber, free from slag or dirt or any coarse crystalline spots. A few crystalline spots may be tolerated, provided that they do not in the aggregate exceed 10 per cent of the sectional area of the bar.

(g) *Cold-bending Test*.—Specimens will be bent cold 180° to an inner diameter equal to the diameter or thickness of the piece tested, except for material $\frac{3}{8}$ inch or under, in which case the inner diameter will be twice the diameter or thickness, and must not show cracks or flaws. When it is not practicable to apply this test to bars or rods in sizes as rolled, test pieces may be prepared $\frac{1}{2}$ by 1 inch, or turned to a diameter of 1 inch.

(h) *Quench-bending Test*.—Specimens will be heated to a temperature of about $1,700^\circ$ F. (yellow heat) and suddenly quenched in water between 78° and 82° F. and bent in the same degree as required for cold bends, and under these conditions must not show cracks or flaws.

(i) *Welding Test*.—Specimens selected from each of the sizes inspected will be worked and welded in the ordinary manner. After welding, specimens will be bent cold across the weld in such manner as to thoroughly demonstrate the welding qualities of the material. Indication of unsoundness at any portion of the weld, unless due to inferior workmanship or causes other than the lack of proper welding qualities, will be cause for the rejection of the material represented by such tests.

(j) *Split Test*.—A rod or bar will be split at the end and each part bent back through an angle of 180° without showing undue extension of the original split or sign of fracture on parts bent back.

(k) *Drift Test*.—A rod or bar will be punched and expanded by drifts until a round hole is formed the diameter of which is not less than nine-tenths the diameter of the rod or width of the bar. Any indications of fracture, cracks, or flaws developed by this test will be sufficient cause for rejection of the lot represented by the rod or bar.

(l) Maximum percentage of phosphorus, 0.15 per cent, and of sulphur, 0.02 per cent.

The table below gives some of the uses to which the various classes of material have been put in the construction of recent vessels. Specific approval for the use of any class of material must be obtained, however, in each case.

Material.	Class.	Use for which suitable.
Engine forgings.....	H. G.	Bolts and studs for all moving parts of main engines, shaft couplings, main bearing caps, thrust bearing side rods, main engine framing, and moving parts of circulating pumps; connecting rods, caps, and bolts; eccentric rods; main circulating pump engine working parts; piston rods; suspension links and link blocks; valve stems.
Engine forgings.....	A	Coupling bolts; crossheads and slippers; crank thrust, line, stern tube, tail, and propeller shafts; main bearing cap bolts; outboard coupling; reverse arms and blocks; rotor shaft; thrust bearing side rods; turning engine worm; working parts, reversing gear; working parts, pumps.
Engine forgings.....	B	Bearer bars; Curtis turbine shaft; engine columns and tie-rods; H. P. relief valve stems; main steam valve stems; main bearing cap bolts; piston rod nuts; piston valve followers; pipe flanges; rotor drum and wheel; stole-plate wedges; swivel pins for crosshead; working levers and gears.
Engine forgings.....	C	Gland for cylinder liners; small parts of eccentrics; uptake and smoke-pipe forgings.
Steel castings.....	A	Engine frame strongbacks; I. P. and L. P. pistons; pistons and followers for piston valves; reverse arms; separators; valve stem crosshead.
Steel castings.....	B	Boiler fittings; crosshead backing guide; cylinder and valve chest covers; engine bedplates; H. P. cylinder relief valves; I. P. and L. P. piston-followers; main bearing caps and shoes; main steam valves; outboard coupling casing; pipe flanges; slip joints, and safety and relief valves for superheated steam; piston rod and valve stem stuffing boxes; reducing valves; reverse shaft bearings; thrust horseshoes; valve stem guide and crosshead cap.
Steel castings.....	C	Unimportant castings.
Iron castings.....		Auxiliary engine cylinders and pistons; crosshead ahead guides; cylinders, cylinder casings, and liners; condensers in main engines; dead plates; distiller shells and flanges; eccentrics; engine bedplates; grate bars; H. P. pistons and followers; low pressure valves and fittings for tugs; Parsons turbine cylinders; pistons and pack-rings; spring bearings; steady bearing caps; stole plates; thrust bearing pedestal.
Iron castings.....	C	Minor castings, as furnace fittings.
Boiler plates.....	A	Boiler plates, not welded or flanged; separators.
Boiler plates.....	B	Boiler drums; boiler plates; evaporator, condenser, and feed water heater shells; welded or flanged boiler plates.
Boiler plates.....	C	Air ducts; ash-pit doors; ash pans; bilge strainers; boiler and superheater casings; blower casings and fans; condenser and feed water heater baffle plates; fair-water sleeve; feed and filter tanks; floor plates; main condenser shapes; oil storage tanks; outboard coupling casing; shapes and plates for uptakes and smoke pipes; other material not essential to structural strength.
Boiler bracing.....	A	{ Boiler bracings as prescribed.
	B	
Rods for bolts, studs, and rivets.	A	Boiler rivets; bolts for outboard couplings; bolts for cylinder and valve chest covers; all bolts not otherwise mentioned securing parts of machinery; all valve bonnets; main bearing cap bolts.
Rods for bolts, studs, and rivets.	B	Bolts in pipe flanges; bolts in valves beneath floor plates connecting rod plugs; rivets for condensers, uptakes, and smoke pipes; structural bolts.
Rods for bolts, studs, and rivets.	C	With boiler plates and shapes, class C.
Boiler tubes.....		Boiler tubes (seamless).
Steam and water pipes (seamless).		All pipes subject to superheated steam; all pipes in engineer department 2 inches and over, unless otherwise directed.
Steam and water pipes (lap-welded).		Bilge suction; dry pipe; link crosshead pin oiling gear; main condenser longitudinal stays; piston valve distance pieces; pipes to reserve feed tanks.
Wrought iron.....		Cap bolt nuts; connecting rod nuts; engine framing nuts; outboard coupling nuts; parts requiring welding; shaft coupling nuts.

G

SPECIFICATIONS FOR INSPECTION OF MATERIAL.

PART II.

Nonferrous Materials for Use in the Construction of Machinery Coming
Under the Cognizance of the Bureau of Steam Engineering,
Navy Department.

(Revised 1910, with Subsequent Modifications.)

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GENERAL INSTRUCTIONS FOR THE INSPECTION OF NONFERROUS MATERIAL.

Coming Under the Cognizance of the Bureau of Steam Engineering,
Navy Department.

DESIGNATION OF CONTRACTOR, MANUFACTURER, AND SUBCONTRACTOR.

1. **Contractor.**—Generally speaking, contractor, as used in these specifications, refers to shipyard, navy yard, or any builder of government machinery, appliances, or structures placing orders for material with some manufacturer.

2. **Manufacturer.**—Refers to person or firm manufacturing material for incorporation in government work being done by shipyard, navy yard, or any other builder of government machinery, appliances, or structures, and who are designated as contractors.

3. **Subcontractor.**—Refers to person or firm to whom the contractor may sublet part of his contract, but not for raw material; the subcontractor in turn places orders with manufacturers for raw material.

OFFICE AND INSPECTORS.

4. **Access to Work and Information.**—The department shall have the right to keep inspectors at the works who shall have free access at all times to all parts thereof and be permitted to examine the raw material and to witness the processes of manufacture.

Contractors and manufacturers shall furnish all the information and facilities the inspector may require for proper inspection under these specifications.

5. **Inspector's Office and Furniture.**—Each firm manufacturing material shall furnish the inspectors, free of expense, with such facilities as may be necessary for the proper transaction of their business as agents of the government.

EXPENSE.

6. **Handling Material.**—All handling of material necessary for purposes of inspection shall be done at the expense of the contractor.

7. **Tests.**—All expense of making tests required by these specifications and the preparation of test specimens, shall be borne by the contractor.

Special appliances necessary to determine that the specification requirements are fulfilled will be furnished by the contractor.

8. **Rejection at Builder's.**—Material may be rejected at navy yards, building yards or works for surface or other defects either existing on arrival or developed in working or storage, even though such material may have previously passed inspection by the government inspector at the place of manufacture. In such cases the contractor must make good any material rejected.

9. **Notification to Manufacturers or Subcontractors.**—Upon receipt of

instructions from the Bureau of Steam Engineering or request from an inspector of machinery, another inspector of material, superintending constructor, or a navy pay office to inspect an order for material, the inspector will at once inform the manufacturers or subcontractors of this fact, and that the material ordered must not be shipped until after inspection, without the inspector's consent. He will also request to be notified when the material will be ready for inspection.

10. Inspection During Process of Manufacture.—The inspector should endeavor to keep in touch with the work throughout its manufacture, and make such efforts as are practicable to secure delivery within contract time.

11. Inspection of Material Without Instructions.—Any material which a manufacturer may present to the naval inspector will be inspected, provided it is without doubt for a bureau whose inspection work is generally done by the Bureau of Steam Engineering, and also provided there is no doubt that the specifications on the company's orders are in accordance with the usual specifications for that class of material.

12. Notifying Bureau of Manufacture of Finished Material Without Orders for Inspection.—The Bureau of Steam Engineering desires to be notified by the naval inspector of all government material being manufactured in his district for Bureau of Steam Engineering upon which no inspection has been ordered. The bureau will then determine whether or not inspection is necessary, if not requested, after the following subparagraphs have been complied with.

Inspectors are authorized to communicate directly with inspectors of machinery, other inspectors of steam engineering material, navy pay offices, or commandants of navy yards, in regard to material that is under inspection, or material the inspection of which has not been requested.

Inspectors of steam engineering material requesting inspection of material of another inspector, will forward the contractor's order in duplicate with signed certificate on face that material ordered is in accordance with approved plans and specifications, to the inspector of material in the district where the material ordered is to be manufactured. A manifold copy of this order will be forwarded to the bureau. Inspectors requesting information in regard to the inspection of material from a navy pay officer will communicate direct with the purchasing pay officer, requesting that duplicate copies of order be forwarded, and that a copy be sent direct to the Bureau of Steam Engineering. A carbon copy of this letter to the purchasing pay officer will be forwarded to the bureau.

ORDERS, LISTS, AND INVOICES.

13. Contractor or Subcontractor will supply the inspector of engineering material or inspector of machinery with the following:

(a) Quadruplicate copies of all orders placed with manufacturers or subcontractors, stating, when possible, the purpose of each item ordered and the specifications for same. On all copies the inspector will certify

that the material is in accordance with approved plans and specifications. One copy will be retained by the inspector, two copies will be sent to the inspector who will inspect the material, and one copy forwarded to the bureau. The copy forwarded to the bureau shall be without letter or indorsement slip. Blue prints of parts will not be forwarded to the bureau unless they give data of the order. The order must also state explicitly what treatment, if any, other than machining, is to be given after leaving the manufacturer.

(b) Triplicate blue prints or sketches when such are necessary in placing orders.

When possible, a separate order should be placed for each vessel or contract.

14. Manufacturer or Subcontractor will furnish the inspector immediately after each shipment:

(a) Copies of each invoice of shipment. This must give:

1. Identification marks.
2. Dimensions of each object.
3. Actual weights of each object or of each lot of similar objects of like dimensions. When material is ordered by weight or gauge, the estimated weights must also be given.
4. Date of shipment.
5. Names of consignor and consignee.
6. Car number and initials of car when shipment is made in carload lots.
7. Number of pieces shipped.

(b) They will also show the inspector all schedules or contractors' orders of material for the government placed with them by the contractors. In case manufacturers or subcontractors make special schedules for their convenience in the conduct of the work, copies of same will, on demand, be supplied to the inspector.

15. The Inspector Will Forward a Copy of Each Shipping Report to the Bureau of Steam Engineering, and also to the naval inspector at the place to which the material is shipped. If material is intended for a navy yard or naval station, the inspector will also forward two copies of each shipping report to the general storekeeper. If the material inspected belongs to another bureau, one copy of shipping report will also be sent to the bureau concerned. Shipping reports will give:

(a) The information required by printed form.

(b) The stamp or stamps which appear on the material inspected, on the casing containing the material or on tags on car in which the material is shipped, and on what part of material, box, or car the stamp is placed.

(c) Date that the manufacturer's report was received stating when the material was or would be ready for inspection.

(d) Date of authorization of shipment.

(e) Initials of assistant inspector that made inspection.

(f) Certificate that copy has been sent to naval representative at destination.

(g) Information of special treatment or inspection.

(h) File number of office copy.

The date of report will be that of date of shipment. Where the manufacturers furnish their own form of report, these may be pasted on the regular form, but all information required will be furnished. This practice should be followed only in exceptional cases.

A shipping report will only cover material supplied on one order. Final report will state the number of shipments made on account of the order.

The report going to consignee shall have the autographic signature of the inspector. Both copies for a general storekeeper shall be so signed; other copies may be signed with stamp.

STAMPS.

16. Identification of Test Coupons and Material from Which Taken.—The coupons, castings, parts of sheet metal, or other material from which tests or analyses are to be taken will be stamped with personal or U S stamp, for identification. This stamp will not be removed by the manufacturer during any part of finishing process, except after duplication by an inspector.

The removal of any inspector's stamp from material without authority of the inspector will be sufficient reason for rejection of that material.

17. Anchor Stamp.—The anchor stamp shall not be put on any material until it has been inspected, weighed, and passed, ready for shipment, and must always appear as evidence of final inspection.

18. Identification Stamp.—The U S stamp, or the inspector's personal inspection stamp, will be used when necessary for identification during the progress of work.

Each object which has passed inspection shall be clearly marked with the necessary stamps, and these stamps on large, rough work shall be encircled by a ring of paint.

19. Stamps on Boxes.—In the case of small articles passed and packed in bulk, or in the case of material which would be injured by stamping, the above-mentioned stamps shall be applied to the boxing or packing material of the object, instead of to the object.

20. Carload Lots—Tags.—If the material is shipped in box cars containing no other freight, it will be sufficient to seal the car and put the stamp on the seal as well as on a tag on the inside of the car near the door.

21. Nonacceptance of Material Without Stamps.—No material will be received at a naval station or building yard for incorporation into naval vessels unless it bears either upon its surface or that of its packing the stamps as evidence that it has passed the required government inspection, as noted on shipping report.

22. Care of Stamps.—Stamps must at all times be carefully guarded by the inspectors and assistant inspectors to whom they may be intrusted, to prevent their use by unauthorized persons.

GENERAL QUALITY.

23. General Character of Material.—All material shall be of domestic manufacture and of uniform quality throughout the mass of each object, and free from all defects.

24. Special Material or Special Treatment.—With the approval of the Bureau of Steam Engineering, special material or special treatment, or both, may be used to obtain the qualities specified.

GENERAL TEST REQUIREMENTS.

25. Tests and Acceptance.—All material for which tests are prescribed shall be inspected and tested by a naval inspector and passed by him, subject to restrictions mentioned herein, before acceptance by the Navy Department.

26. Treatment of Test Pieces.—Test pieces, after being cut from the plate or object to be tested, shall not be subjected to any treatment or process except machining to size; and such pieces shall not be cut off until the plate or object shall have received final treatment, except in those special cases mentioned in the following specifications.

27. Flaws in Test Pieces.—Test pieces which show defective machining or which, after breaking, show flaws, or which break outside of the measuring points, may be discarded, and the inspector will select others in their stead.

28. Test Pieces for Lots.—Test pieces which represent groups or lots shall be taken, as nearly as the case will permit, so as to represent the worst material in that lot.

29. Location of Test Pieces.—All test pieces of rolled bars which are too large to be pulled in their full sizes shall, unless otherwise specified, be taken at a distance from the longitudinal axis of the object equal to one-quarter of the greatest transverse dimension of the body of the object, not including palms and flanges.

The test pieces should be taken from a part of the material which, with the exception of palms or flanges, has not been reduced by forging or rolling more than any other part of that piece of material.

CHEMICAL ANALYSIS.

30. Government's Analysis.—Drillings for analysis must be fine, clean, dry, and free from scale and iron drillings or filings. The inspector may take them from any test piece or from any part of the material, provided in this last case that by so doing the material will not be rendered unfit for use. For valves under 4 inches, and all screw fittings, one article will be taken at random from each lot of the same dimensions. If the chemical composition be not found as required, the entire lot will be rejected. A lot is not to exceed 400 pounds by weight. Unless otherwise

directed, the chemist will make determinations of those elements only which are limited by the specifications. All samples will be freed from iron particles by a suitable magnetic separator. Samples for chemical analysis will be taken as follows:

From rods, tubes, etc., material from the ends of four rods for lots of 400 pounds of rods and from the test tubes for the lots of tubes prescribed in article 82, shall be taken across the entire cross section, of about 1 ounce each. These samples to be thoroughly blended, one-half for the manufacturer and one-half for the inspector, the amount retained by the inspector to be sufficient for three analyses. Any question of the results given in the first report of analysis will be decided by the results of an additional and third analysis from the remainder of the samples. Reports of failure will only be made after the original and check analyses have shown that the material does not meet the specifications.

Samples from other materials will be taken on the same principle, plates and shapes being drilled entirely through the material.

Cuttings for analysis should be prepared as follows:

Cuttings should be made by a milling machine or a "shaper" set to cut chips the size of a medium pin's head. In case neither is available a drill may be used, which should be ground flat and not pushed.

In case the samples represent a large amount of material, or the material is urgently needed to carry on work, the order for the analysis should be marked "Special" and reasons given for so doing.

All metals of a proprietary nature will be subjected to chemical analysis, and will be accepted only after approval by the bureau, if they differ from the specifications of standard mixtures.

ADDITIONAL TESTS.

31. By Bureau's Orders.—Tests may be prescribed by the Bureau of Steam Engineering for the inspection of material for which tests are not specified herein.

This shall have reference to all special or proprietary mixtures, such as special antifriction metals; metals to be run under the extrusion process, and all brasses or bronzes cast or rolled, which are intended to take the place of the standard mixtures particularized in these specifications.

32. By Inspector's Decision.—The inspector may make, from time to time, such additional tests as he may deem necessary to determine the uniformity of the material.

TENSILE TESTS AND TEST PIECES.

33. Interpretation.—The tensile strength herein specified means the ultimate tensile strength per square inch of original cross section. The yield point may be measured by the drop of the beam or the halt of the gauge of the testing machine. The elongation is that obtained after fracture. In the case of test pieces of rectangular section the reduction

of area is to be measured by the product of the average width and thickness of the reduced area, and not the minimum width and thickness.

34. Pulling Speed.—Each tensile-test piece shall be subjected to a direct tensile stress until it breaks in a machine of standard manufacture, running at a pulling speed of not less than 1 inch and not more than 5 inches per minute for 8-inch test pieces and not less than $\frac{1}{2}$ inch and not more than three inches per minute for 2-inch pieces.

35. Uniformity of Section.—Tensile-test pieces shall be uniform in cross section between measuring points.

36. Allowance of Variation in Area of Test Pieces.—A variation of 5 per cent above or below in area is allowed.

37. Standard Area and Length.—Test pieces from castings of metal, the required tensile strength of which is below 40,000 pounds, are to have a length of 2 inches between measuring points and an area of cross section of 1 square inch. Other tensile-test pieces will be the standard test piece of 0.505 inch diameter and 2 inches between measuring points.

38. Full-size Bars.—Full-size bars and rods within the capacity of the testing machine may be used as tensile-test pieces, and in this case the bending tests may also be taken from the full-size bars and rods.

In the case of bars or rods too large to be tested in full size, the tensile-test pieces will be taken from a point midway between the center and the circumference.

When the full-sized bar used as a tensile test is less than $\frac{1}{2}$ inch in diameter, or the effective diameter is less than $\frac{1}{2}$ inch, the distance between measuring points will be four times the diameter or the effective diameter. The latter will be taken as the mean of the width and thickness.

39. Plates, Standard Width for Test Pieces.—The standard width of tensile-test pieces from plates and tubes will be $1\frac{1}{2}$ inches, the thickness the same as the plate or tube, and the length between measuring points 8 inches.

40. Rounding of Edges of Test Pieces.—In the case of bending-test pieces of rectangular section, the edges may be rounded off to a radius equal to one-fourth of the thickness.

41. Standard.—Bending-test pieces shall be 1 inch wide by $\frac{1}{2}$ inch thick. For plates the width shall be not less than $1\frac{1}{2}$ inches and the thickness that of the plate. The bending may be done by either pressure or by blows.

SPECIFICATIONS FOR NONFERROUS CASTING MATERIALS.

42. The inspection of these castings shall conform to the "General instructions for the inspection of nonferrous material coming under the cognizance of the Bureau of Steam Engineering."

43. The composition must be made of such materials as will give the required chemical analysis. Scrap will not be used except such as may result from the process of manufacture of articles of similar composition.

Letter.	Name.	Composition by percentage.					
		Copper.	Tin.	Zinc.	Iron, maximum.	Lead, maximum.	Miscellaneous.
B-c....	Commercial brass	88 64-68	10	2 32-34	2.0	3.0	
D-c....	Muntz metal.....	59-62		39-41		.6	
F.....	Brazing metal.....	84-86		Rem.	.06	.3	
G.....	Gun bronze.....	87-89	9-11	1-3	.06	.2	
→ H.....	Journal bronze	82-84	12.5-14.5	2.5-4.5	.06	1.0	
M.....	Valve bronze.....	87	7	Rem.	.06	1.0	
Mn-c...	Manganese bronze ¹ ...	57-60	.75	37-40	1.0		Aluminum, 0.5; manganese, 0.3.
Mo-c...	Monel metal.....	Rem.			6.5	.0	Aluminum, 0.5; nickel, 60 (min.).
N-c....	Cast naval brass.....	59-63	.5-1.5	Rem.	.06	.6	
P-c....	Phosphor-bronze ¹	80-90	6-8	Rem.	.06	.2	Phosphorus, 0.3.
S-c....	Screw pipe fittings, brass.	77-80	4	13-19	.1	3.0	
Ni.....	Nickel.....						Nickel, 97 (min.).
Cu.....	Copper.....	99.8 (Min.)					To be free from sulphur and other impurities and metals.
Zn.....	Zinc (rolled plates for boilers or slabs).			98½ (Min.)	.08		
Sn.....	Tin.....		99.6 (Min.)				
Pb.....	Lead (1).....					Min. 99½	
	Lead (2).....					97½	
Me-c...	Monel metal (ingots or shot).				3.0		Nickel, 60 per cent (min.); manganese, 2 per cent (min.); balance copper, with small amounts of other ingredients not injurious to casting qualities, or detrimental to strength or noncorrosive properties.

¹ The proportions given are approximate and are a guide as to about the proportions of the elements that are expected, except for lead, aluminum, and iron, which are maximum limits.

45. The following physical requirements will be met by castings not for ornamental purposes, of G, Mn-c, Mo-c, and P-c:

The physical tests may be waived by the bureau or by the inspector requesting inspection on small castings of which the factor of safety is large by reason of necessities of design.

	G.	Mn-c.	Mo-c.	P-c.
Tensile (minimum).....	30,000	60,000	65,000	40,000
Yield point (minimum).....	15,000	30,000	32,500	20,000
Elongation in 2-inch (minimum).....per cent..	15	20	25	20

46. The castings must be made in accordance with the drawings and specifications—sound, clean, free from blowholes, porous places, cracks, or any other defects which will materially affect their strength or appearance or which indicate an inferior quality of metal.

47. In the case of screw propellers of manganese bronze, coupons will be cast attached to the hub and to each blade; the coupons will be cast flat on the blade or hub; those on the blades will be attached at half the distance from the root to the periphery. The coupons will be cast of such size as may be found necessary to give a sound test specimen, and turned down as required by paragraph 37. The coupons are to have no treatment other than machining to reduce them to the proper diameter. For castings weighing over 200 pounds test pieces or coupons shall be taken in such number and from such parts of the casting as will thoroughly exhibit the quality of the metal, not less than three being taken for large propeller blades. For monel metal propellers, test coupons will be attached to the hub and will be three in number. Two will be tested and one retained for test in case of defective specimens.

48. Castings of G, Mn-c, Mo-c, or P-c, weighing less than 200 pounds finished, may be tested by lots or heat, a lot not to exceed 200 pounds, and heat not to exceed 500 pounds of finished castings. Each lot or heat will be represented by one test specimen when attached to a casting or when a casting is sacrificed to obtain a test specimen. If the castings are too small for the attachment of coupons, the test pieces may be cast separately from the same metal under as nearly as possible the same condition as the casting. Where test pieces are cast separately from the castings, two test pieces will be required, one to be poured before and one after the castings.

49. Coupons shall not be detached from castings until they are stamped by the inspector. If the test pieces are cast separately from the casting they must be cast in the same flask with the casting and must be received from it in the presence of the inspector and stamped by him at the time they are taken out of the molds.

50. The color of the fractured section of the test pieces and the grain of the metal must be uniform throughout.

SPECIFICATIONS FOR ROLLED PLATES, SHEETS, ROUNDS, AND SHAPES; OF NONFERROUS MATERIALS.

51. The inspection must conform to the "General instructions for the inspection of material coming under the cognizance of the Bureau of Steam Engineering."

52. The composition must be made of such materials as will give the required chemical analysis. Scrap will not be used except such as may result from the process of manufacture of articles of similar composition.

53.

Letter.	Name.	Composition by percentage.					
		Copper.	Tin.	Zinc.	Lead, maximum.	Iron, maximum.	Miscellaneous.
A	Admiralty	70 (min.)	1 (min.)	Rem.	0.075	0.06	
Be-r....	Benedict nickel....	84-86					Rem. nickel.
B-r....	Sheet brass and piping.	60-70		Rem.	.5		
B-r....	Commercial brass rod.	60-63		Rem.	3.0		
Cu-r....	Copper	99.5 (min.)					
D-r....	Muntz metal.....	59-62		Rem.	.6		
P-r....	Phosphor bronze. ¹	85-95	5-10	⁴ (max.)	.2	.06	Phosphorus, 0.15.
Mn-r....	Manganese bronze. ¹	57-60	0.5	37-40		1.0	Manganese, 0.30.
Mo-r....	Monel metal.....	Rem.			.0	3.5	Nickel, 60 (min.); aluminum, 0.5 (max.).
N-r....	Rolled naval brass.	59-63	.5-1.5	Rem.	.2	.06	

¹The proportions given are approximate and are a guide as to about the proportions of the elements that are expected, except for lead, aluminum and iron, which are maximum limits.

54. Test specimens will show the following results:

Letter.	Name.	Ultimate tensile strength per square inch (minimum).	Yield point (minimum).	Elongation in 2 inches (minimum).
				<i>Per cent.</i>
Cu-r ...	Copper.....	30,000		25
D-r ...	Muntz.....	40,000	$\frac{1}{2}$ ultimate.	25
P-r ...	Phosphor bronze.....	50,000	do.....	25
N-r ...	Naval brass, 1 inch and below.....	62,000	do.....	25
	Above 1 inch.....	60,000	do.....	28
Mn-r....	Manganese, 1 inch and below.....	72,000	do.....	28
	Above 1 inch.....	70,000	do.....	30
Mo-r....	Monel metal, 1 inch and below	84,000	47,000	25
	Above 1 inch.....	80,000	45,000	28

55. Test pieces are to be as nearly as possible of the same diameter as the rounds, or else they are to be not less than $\frac{1}{2}$ inch in diameter and taken at a distance from the circumference equal to one-half the radius of the rounds.

56. Test specimens for rounds and bars of N-r, Mn-r, Mo-r, will stand:

- (a) Being hammered hot to a fine point.
- (b) Being bent cold through an angle of 120° and to a radius equal to the diameter or thickness of the bars.
- (c) The bending bar may be the full-sized bar or the standard bar of 1 inch width and $\frac{1}{2}$ inch thickness. In the case of bending test pieces of rectangular section, the edges may be rounded off to a radius equal to one-fourth of the thickness.

Fractures of specimens must show throughout uniform color and grain.

57. Various composition materials, otherwise conforming to the specifications, but manufactured under proprietary processes or having proprietary names, will be accepted as rolled naval brass provided the ingredients are approved by the bureau.

58. Surface Inspection.—The plates, sheets, rounds, and shapes must be free from all surface defects, clean and straight, of uniform color, quality, and gauge. In calculating the weight of sheets and plates, the following unit weights shall be taken: The weight of 1 cubic inch of hot-rolled copper to be 0.320 pound; 1 cubic inch of cold-rolled copper, 0.323 pound; 1 cubic inch of rolled Muntz metal, 0.296 to 0.313 pound, according to its composition; monel metal, 0.323 pound.

59. Sheets and Plates will be cut to the required dimensions. Material shall be ordered in as narrow widths as can be used and in stock lengths, except for a particular job where ordering in stock lengths would result in waste, when exact lengths will be specified.

60. Tolerance for Weight and Gauge.—No excess weight of the lot will be paid for, and no single piece that weighs more than 5 per cent above the calculated weight will be accepted.

The tolerance for underweight and gauge will be 5 per cent for plates up to and including 48 inches wide, 7 per cent for plates above 48 to and including 60 inches wide, and 8 per cent for plates over 60 inches wide.

61. The following will be considered stock lengths for brass sheets when ordered to 10-foot lengths:

Forty per cent in weight may be in 8 to 10 foot lengths.

Thirty per cent in weight may be in 6 to 8 foot lengths.

Twenty per cent in weight may be in 4 to 6 foot lengths.

Ten per cent in weight may be in 2 to 4 foot lengths.

No lengths less than 2 feet will be accepted, and the total weight of all pieces on lengths less than 10 feet must not exceed 40 per cent in any one shipment.

62. In general, the terms used in giving the temper will be soft, medium hard, hard, extra hard. Other trade names to designate the temper are dead soft, half hard, three-quarters hard.

63. Spring brass shall not be ordered in widths exceeding 12 inches.

64. Rods, Bars, and Shapes.—When ordered to any length, the material will be accepted in stock lengths unless it is specifically stated that the lengths are to be exact. Stock lengths will be as follows:

When ordered in 12-foot lengths, no length less than 8 feet.

When ordered in 10-foot lengths, no length less than 6 feet.

When ordered in 8-foot lengths, no length less than 6 feet.

When ordered in 6-foot lengths, no length less than 4 feet.

When ordered to the lengths given above, the weight of lengths less than length ordered shall not exceed 40 per cent of any one shipment.

This applies to all rods from $\frac{1}{4}$ inch to 1 inch in diameter or thickness, whether round, rectangular, square, or hexagonal.

Above 1 inch, to and including 2 inches, random lengths from 4 to 10 feet. Above 2 inches no length will be less than 4 feet.

65. Spring Wire.—Will be elastic and no harder than will meet the temper required.

**SPECIFICATIONS FOR SEAMLESS PIPE, BRASS (IRON PIPE SIZES),
COPPER, BENEDICT NICKEL, MONEL METAL, OR OTHER EQUIVA-
LENT METAL—MATERIAL.**

66. Chemical analysis must show the material to be in accord with the composition required by table, paragraph 53, for rolled metals of name given.

When it is specified that the pipe is to be threaded, the threading is to be in accordance with the Briggs formula for pipe threads, which has been adopted as the standard pipe thread for use in the United States Navy.

67. The pipe must be inspected for surface defects, and it must be free from cracks, seams, and defects generally.

68. Each pipe must withstand an internal hydraulic pressure which will subject the metal to a stress of:

s =	B-r	7,000
	Cu	6,000
	Be-r	14,000
	Mo-r	20,000 lbs.—

per square inch, in accordance with the formula for thin, hollow cylinders under tension, where

$$p = \frac{2ts}{d}$$

p = Safe internal pressure,

d = Inside diameter of pipe in inches,

s = Safe tensile strength of material given above,

t = Thickness of pipe in inches,

but brass and copper pipe will not be tested beyond 1000 pounds nor Benedict and monel metal pipes beyond 2000 pounds per square inch per gauge, unless specially directed. Every pipe must be perfectly tight under pressure and show no signs of bulging, cracks, flaws, porous places, or other defects.

SPECIAL FOR BRASS PIPE.

69. All pipe, unless ordered "hard," is to be annealed sufficiently to prevent fire cracking and to stand the physical tests.

70. When the pipe is finished (ready for shipment), the naval inspector will subject 1 per cent of the lot, taken at random, to the following physical tests:

(a) The end of each test pipe must stand being flattened by hammering until the sides are brought parallel with a curve on the inside at the ends not greater in diameter than twice the thickness of the metal in the pipe, without showing cracks or flaws.

(b) Each test pipe shall have a piece 3 inches long cut from it, which piece when split must stand opening out flat without showing cracks or flaws.

(c) Each test pipe must stand threading in a satisfactory manner with the usual thread for the size of the pipe. When the pipe is ordered "semiannealed," the (a) and (b) tests shall be made on annealed test specimens. These (a), (b), (c) tests shall be made on each of the test pipes, and the test specimens shall be furnished at the contractor's expense. If any of the pipes selected for test fail, the naval inspector will select two extra pipes from the same lot and put them through the same test as the pipe that failed, and both of these pipes must be found satisfactory in order that the lot may be passed. The failure to pass satisfactorily any one of the tests marked (a), (b), (c), will reject the lot.

Brass pipe for radiators for heating system to be semiannealed, and to stand bending 180° around a diameter of $1\frac{1}{2}$ times the outside diameter of the pipe without cracking.

71. The standard weight for seamless-drawn brass pipe will be 0.307 pound per cubic inch of material. No excess weight in the entire lot will be paid for. No single piece that exceeds calculated weight by more than 5 per cent will be accepted. A tolerance of 5 per cent under weight and gauge will be allowed on pipes up to and including 6 inches I. D., 7 per cent on pipes from 6 inches up to and including 8 inches I. D., and 8 per cent on pipes above 8 inches I. D. Where ordered by the number of feet, commercial lengths will be accepted, viz, from 10 to 16 feet. Each large single pipe, or bundle of small pipes, must be marked with the name of the vessel for which it is intended or with the number of the contract, bureau order, or requisition.

**SPECIAL FOR BENEDICT NICKEL, MONEL METAL, OR OTHER
EQUIVALENT METAL PIPE.**

72. When the pipe is finished (ready for shipment), the naval inspector will subject 1 per cent of the pipes, taken at random, to the following physical tests:

(a) Test specimens selected from these shall not show less than 80,000 pounds tensile strength per square inch for monel metal and 60,000 pounds for Benedict metal, hard drawn.

(b) Each test pipe must stand being flattened along different elements by hammering until the sides are brought parallel, with a curve on the inside at the ends not greater in diameter than the thickness of the metal in the pipe, without showing cracks or flaws.

(c) Each test pipe shall have a piece 3 inches long cut from it, which piece when split must stand opening out flat without showing cracks or flaws.

(d) Each test pipe must stand threading in a satisfactory manner with the usual thread for the size of the pipe.

(e) The end of a pipe, cold, must stand having a taper pin, taper $1\frac{1}{2}$ inches to the foot, driven into it until the end of the piece stretches to $1\frac{1}{8}$ times the original diameter without showing cracks or flaws.

(f) A piece of the pipe one diameter long must stand crushing in the direction of its axis under a hammer until shortened to three times the thickness of the pipe in height without showing cracks or flaws.

(g) A piece of pipe must stand flanging cold, the width of the flanging to be one-fourth of the inside diameter of the pipe.

(h) *Hot or Fire-crack Test.*—Pieces 2 feet long, cut from each test specimen, shall be heated to about 300° F., and must then stand plunging into ice-cold water without showing cracks.

(i) *Cold-bending Tests.*—Each test specimen, cold, shall be sawed through the axis for a distance of 1 foot, then the split parts folded back across the grain, flat on themselves, without showing fracture.

These tests shall be made on each of the 1 per cent test pipes, and the test specimens shall be furnished at the contractor's expense.

If any one of these pipes selected for test fails, the naval inspector will select two extra pipes from the same lot and put them through the same test as the pipe that failed, and both of these pipes must be found satisfactory in order that the lot may be passed. The failure to pass satisfactorily any one of the tests will reject the lot.

73. Each large single pipe or bundle of small pipes will be marked with the name of the vessel for which it is intended, or with the number of the contract, bureau order, or requisition. The standard weight for seamless "Benedict nickel" pipe will be 0.323 pound per cubic inch of material, and for monel metal, 0.323 pound.

SPECIAL FOR COPPER PIPES.

74. The pipe must be free from indentation, cracks, flaws, or other surface defects, inside and outside, perfectly round, of the specified diameter. All straight sections of pipe 10 inches in diameter (inside) or less shall be seamless drawn.

75. A strip $1\frac{1}{2}$ inches wide will be taken from each lot of 2,000 pounds or less of pipe, and must stand the following tests:

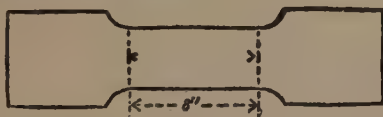
(a) If less than half an inch thick, it must stand bending flat back cold after being annealed.

(b) If half an inch or over, it must bend back after being annealed until the ends are parallel and the inner radius of the bend is equal to the thickness of the piece.

(c) In every case the ends of the bending test pieces shall stand hammering down hot to a knife edge without showing signs of cracks.

(d) The piping from which test pieces are taken must stand flanging cold, the width of the flanging to be one-fourth of the inside diameter of the pipe after annealing.

76. Pipes of 2 inches inside diameter and over, for steam or feed pipes or other such high pressures, are to be subject to tensile tests, one piece of pipe from each lot of 1000 pounds or less being selected to represent the lot. If the pipes are from 2 inches to 6 inches inside diameter, the test pieces are to be cut longitudinally. If over 6 inches inside diameter, they will be cut circumferentially. The test pieces will be heated to a cherry red and straightened when hot, then machined to the shape shown in the sketch below, care being taken to have the brazed seam, if any, between the measuring points.



77. For thicknesses up to and including $\frac{1}{4}$ inch, the width of the narrow part of the test piece shall be about $1\frac{1}{2}$ inches. For thicker pieces the width shall be such as to give a cross section of about half a square inch, but the breadth shall not in any case be less than the thickness. The drawn surfaces are not to be machined, but to be left in their original condition.

78. The test piece must show an ultimate tensile strength after being annealed of at least 28,000 pounds per square inch for all pipe, and an elongation of at least 25 per cent in 8 inches in the case of seamless pipe.

79. Thickness.—Tolerance for under gauge will be the same as for underweight.

80. Weight.—No excess of weight of lot of pipe above the calculated weight, on a basis of 1 cubic inch of copper pipe weighing 0.323 pound, will be paid for. No single piece that weighs more than 5 per cent above the calculated weight will be accepted. A tolerance of underweight equal

to 5 per cent of the calculated weight will be allowed for seamless pipe up to and including 6 inches inside diameter, from 6 inches to and including 8 inches I. D. a tolerance of 7 per cent underweight for 8 inches I. D. and above a tolerance of 8 per cent underweight.

SPECIFICATIONS FOR SEAMLESS TUBES FOR CONDENSERS, DISTILLERS, AND FEED WATER HEATERS.

81. Material.—The tubes may be made of one of the following metals, shown in table, paragraph 53: Admiralty, Benedict, monel. Chemical analysis must show the material to be in accord with that table. No scrap will be used in the manufacture, save such clean scrap as may result from process of manufacture of similar articles. The order for the tubes will state the material. Distiller tubes will be tinned.

82. Inspection.—The tubes of each lot will be assembled for inspection in a well-lighted place and so arranged as to simplify the inspection as much as practicable. Tubes that do not show uniform annealing under handling will be rejected from the lot. For purpose of inspection, the tubes will be divided into lots of 600, plus 1 per cent required for test purposes. The latter will be exclusive of the number required on the order and will be furnished at the contractor's expense.

83. Surface Inspection.—All tubes must be seamless, true to form, of an equal thickness throughout, of workmanlike finish, free from cracks, seams, and other defects, and stiff enough to lie straight when resting on supports 6 feet apart. If 10 per cent of the tubes of any lot show slivers on the inside, the lot will be rejected.

84. Weight.—The weight of the test tubes of any lot will be used in calculating the weight of the lot. All tubes must be up to the required gauge on the thinnest side. An excess of weight of not more than 5 per cent will be allowed on all tubes. Tubes of Admiralty mixture must weigh not less than 0.308 pound per cubic inch of metal.

Benedict nickel tubes must weigh not less than 0.323 pound per cubic inch of metal.

Monel metal tubes shall weigh not less than 0.323 pound per cubic inch.

85. Tests.—(a) Samples cut from the ends of tubes selected for tests will be subjected to a compression test in a standard testing machine under the following conditions: The sample will be held between two blocks, giving contact on opposite sides of the tube for a distance of an inch at the end. The pressure to be applied gradually until the required pressure for the size and gauge is reached, when the pressure will be released and the resulting compression between the points of contact must not exceed 0.005 inch, at the end of the tube.

Pressure to be applied:	Pounds.
No. 20 B. W. G., $\frac{5}{8}$ -inch O. D.....	200
No. 18 B. W. G., $\frac{3}{4}$ -inch O. D.....	350
No. 16 B. W. G., $\frac{7}{8}$ -inch O. D.....	550
No. 14 B. W. G., $\frac{1}{2}$ -inch O. D.....	950
No. 16 B. W. G., $\frac{3}{4}$ -inch O. D.....	450

(b) Test tubes will be hammered with a hand hammer, flat on different elements, throughout balance of length remaining after sections for expanding and compression test in paragraph (a) are cut off. The tubes will not show signs of stress under this test. The term "hammered flat" will be interpreted as follows: A micrometer caliper set at three times the thickness must pass over the tube freely throughout flattened part except at the point where change in element of flattening takes place.

(c) Each test tube will be subjected to an expanding test. This will be carried out by driving a pin with a taper of $\frac{3}{4}$ inch to the foot into the tube until the internal diameter has been increased $16\frac{2}{3}$ per cent.

(d) All tubes will be subjected to a hydrostatic pressure of 1000 pounds per square inch, and then to a pneumatic pressure of at least 80 pounds per square inch, the tubes being immersed in water during application of the latter pressure.

86. The failure of more than one test tube to stand each test, or the same tube failing in more than one test, will cause the rejection of the lot. In the case of the failure of one tube under a single test, two other tubes will be subjected to the test, and both tubes must stand the same test; the failure of one will cause rejection of the lot.

87. Tinning and Annealing.—All tinned tubes must be acid cleaned, dipped in molten tin, and then immediately wiped with hempen tow inside and out to insure their being smooth on both surfaces. All tubes of 70 per cent copper and 29 per cent zinc and 1 per cent tin must be annealed and acid cleaned. Benedict nickel tubes shall not be annealed.

The amount of annealing required for condenser tubes shall be sufficient to enable the tubes to pass the physical and compression tests of paragraph 85.

88. Tubes for feed water heaters and gland stock, and such other tubes (except Benedict nickel) as are larger and thicker than condenser tubes, must be sufficiently annealed to prevent cracking.

89. Benedict nickel and monel metal tubes are distinguished from brass by their greater hardness and density and by their color, which is like that of tin, and is uniform throughout the material.

90. Condenser tubes will be packed and cased in lots as follows

$\frac{5}{8}$ No. 18, $12\frac{1}{2}$ to 15 feet.....	100
10 to $12\frac{1}{2}$ feet	120
$7\frac{1}{2}$ to 10 feet.....	150
6 to $7\frac{1}{2}$ feet.....	200
$\frac{5}{8}$ No. 16, 13 to 15 feet.....	60
10 to 13 feet.....	75
$7\frac{1}{2}$ to 10 feet.....	100
6 to $7\frac{1}{2}$ feet.....	120
$\frac{3}{4}$ No. 14, 14 feet.....	60
$\frac{7}{8}$ No. 14, 21 feet 10 inches.....	30

91. Each case will have stenciled on it in addition to the usual shipping information the "name of ship" or the number of contract; the number and dimensions of the tubes; the name of the manufacturer. The packing will be solid enough to stand transportation and handling without injury to the tubes.

SPECIFICATIONS FOR WHITE METAL.

92. **Material.**—The composition shall consist of 3.7 per cent of the best refined copper, 88.8 per cent Banca tin, 7.5 per cent of the regulus of antimony, and shall be well fluxed with borax and rosin in mixing.

93. When practicable, the weighing and mixing of the metals will be witnessed by a government inspector. Otherwise as many chemical analyses will be taken as, in the judgment of the naval inspector, will show that the material is of the proper composition.

94. If by reason of scarcity Banca tin cannot be procured, another standard brand of tin may be proposed, subject to the approval of the Bureau of Steam Engineering.

SPECIFICATIONS FOR TURBINE MATERIALS.

95. **Blading and binding strips** may be cold rolled and drawn, or extruded and rolled. May be of the following metals:

Monel metal.

Composition: 71-73 per cent copper; lead not to exceed 0.2 per cent; balance zinc.

Cupro-nickel: 79-81 per cent copper; iron not to exceed 0.75 per cent; balance nickel.

Composition: 53-56 per cent copper; 1.5 per cent nickel alloy; lead not to exceed 0.5 per cent; balance zinc.

Phosphor-bronze: As given in paragraph 53.

96. The blades or strips will be hard, smooth, straight, and free from imperfections, and true to gauge. Maximum and minimum gauges will be furnished for testing the blading.

97. **Calking Strips.**—Composition: 62-65 per cent copper; lead not to exceed 0.6 per cent; balance zinc. Calking strips will be annealed soft.

98. **Dummy and Gland Strips.**—Naval rolled brass or composition: 71-73 per cent copper; lead not to exceed 0.2 per cent; balance zinc.

To be rolled or drawn hard.

99. **Thrust Rings.**—Composition: 82 to 84 per cent copper; 12.5 to 14.5 per cent tin; 2.5 to 4.5 per cent lead.

100. The following table gives the uses for which the various classes of material are suitable.

Letter.	Name.	Purposes for which suitable.
B-c....	Commercial brass.	Name and number of plates; cases for instruments; oil cups; distributing boxes.
Cu-r....	Copper.....	Copper pipe and tubing.
D-c....	Muntz metal.....	Brazing metal, and all flanges and fittings that are to be brazed. All composition valves 4 inches in diameter and above; expansion joints, flanged pipe fittings, gear wheels, bolts and nuts, miscellaneous brass castings, all parts where strength is required of brass castings or where subjected to salt water, and for all purposes where no other alloy is specified. Composition valves: Safety and relief, feed check and stop, surface blow, drain, air, and water cocks, main stop, throttle, reducing, sea, safety sluice, and manifolds at pumps. Condenser..... Distiller..... Feed-water heater..... Oil cooler..... Heads, shapes, and water chests. Pumps: Air-pump, casing, valve seats, buckets, main circulating, water cylinders, valve boxes, water pistons, stuffing boxes, followers, glands, in general the water end of pumps complete except as specified. Stuffing boxes: Glands, bushings for iron or steel boxes. Blowers: Bearing boxes. Journal boxes: Distance pieces. Miscellaneous: Grease extractors, steam strainers, separators, casing for stern tube and propeller shafts, propeller hub caps. Bearings: Main, stern tube, strut and spring. Spring bearings: Glands and baffles. Reciprocating engine: Intermediate and low pressure relief valves and casings, crosshead brasses, crank pin brasses, eccentric straps and distance pieces.
F.....	Brazing metal.....	
G.....	Gun bronze.....	
H.....	Journal bronze...	Journal boxes, guide gibs, bushings, sleeves, slippers, etc. Reciprocating engine: Valve stem crosshead bottom brass; link block gibs, suspension link brasses.
M.....	Valve bronze.....	All valves below 4 inches in diameter, for steam and general purposes, for which the material is not otherwise specified; hose couplings and fittings.
Mn-c....	Manganese bronze	Propeller hubs, blades, engine framing, and composition castings requiring great strength.
Mo-c....	Monel metal.....	Same as for Mn-c, and pump liners, valve seats, and castings requiring great strength and hardness and incorrodibility; shaft nuts and caps.
N-c....	Cast naval brass..	Valve handwheels, hand-rail fittings, ornamental and miscellaneous castings, and valves in water chests of condensers.
P-c....	Phosphor bronze..	Castings where strength and incorrodibility are required.
S-c....	Screw pipe fittings	For composition screwed fittings.
Ni.....	Nickel.....	For valve seats.
A.....	Admiralty metal..	Condenser, distiller, feed-water heater, and evaporator tubes.
B-n....	Benedict metal....	Do.
D-r....	Muntz metal.....	Bolts and nuts not subject to action of salt water.
P-r....	Phosphor bronze..	Pump rods, valve stems, valve springs, etc., exposed to salt water.
Mn-r....	Manganese bronze	Rolled rounds, used principally for propeller blade bolts, air pump and condenser bolts, and parts requiring strength and incorrodibility.
Mo-r....	Monel metal.....	Rolled rounds, used principally for propeller blade bolts, air pump and condenser bolts, and parts requiring strength and incorrodibility, and pump rods.
N-r....	Rolled naval brass	Rolled rounds, used principally for propeller blade bolts, air pump condenser bolts, and parts requiring strength and incorrodibility, and pump rods, tube sheets, supporting plates, and shaft for valves in water heads for condensers.
B-r....	Commercial rolled brass.	Sheet brass: For liners, trim, etc.
		Brass pipe: Boiler dry pipe, hand rails.
		Distributing oil tubes and water pipes.
		Commercial brass rod for trim, and purposes where strength and incorrodibility are not required.
W.....	Antifriction metal	Lining bearings.
Cu.....	Copper.....	Ingots.
Zn.....	Zinc.....	Slabs.
		Rolled.
Sn.....	Tin.....	Ingots.
Pb.....	Lead.....	Figs.

H

In order to determine the fitness of all line officers in engineering the following questions have been prepared by the Navy Department as an aid and guidance for examining boards in their examination of officers below the grade of lieutenant-commander.

The portion of the book referring to these questions is indicated below each question.

EXAMINATION QUESTIONS FOR LINE OFFICERS.

SUBJECT NO. 1.—GENERAL KNOWLEDGE OF MACHINERY.

Subdivision A.—Marine Engines.

1. Describe a vertical inverted triple expansion marine engine, giving material of which different parts are made and the functions of these parts. Page 47.
2. Describe briefly the engines used in modern war vessels as regards their position and arrangement of cylinders. Page 125.
3. Describe briefly the types of engines used in modern war vessels from the standpoint of the way in which the steam is expanded. Page 120.
4. Name the attachments to the engines of modern war vessels and describe three (to be selected by the board). Chapters VII and VIII.
5. Sketch and describe a steam piston valve and an ordinary steam slide valve. Explain the advantages and disadvantages of these two types. Chapter IV.
6. Describe the Stephenson valve gear. Page 94.
7. How are multiple expansion marine engines fitted in order to permit of varying the amount of cut-off of the steam in each cylinder at will? Page 100.
8. Describe the arrangement of valve gear fitted to a reversing engine, by which the amount of travel of the valve link is automatically controlled. Page 102.
9. Describe the principal features of a surface condenser. Name and describe the functions of the different pumps connected to a surface condenser. Chapter IX.
10. Draw a good practical indicator card as obtained from an engine in good condition, and show by dotted lines the results of defects in adjustment or fitting of the valve and other parts. Page 232.

Subdivision B.—Marine Boilers.

1. Give a brief description of a Scotch boiler, explaining the functions of the different parts.
2. Describe a modern marine Babcock & Wilcox boiler.
3. Describe a Niclausse water tube boiler.
4. Describe any one of the small tube water tube boilers in common use for torpedo boats and torpedo-boat destroyers.
5. Give a brief explanation of what is meant by the terms "Closed fire room systems of forced draft" and "Closed ash pit system of forced draft," with the comparative advantages and disadvantages of the two systems.
6. Name the various attachments to a marine boiler and describe three (to be selected by the board).
7. Give the relative advantages and disadvantages of shell and water tube boilers.
8. What is meant by the term "priming" of a boiler, and what are the causes tending to produce it?
9. What is meant by the terms "heating surface," "calorimeter," "grate surface?" What are the approximate ratios of heating surface to grate surface in water tube boilers and in Scotch boilers?
10. Into what three general classes, according to their water circulation, may water tube boilers be divided? Give an example of each class.

Subdivision C.—Auxiliary Machinery and Hull Drainage.

1. Describe a boiler feed pump for high pressures, giving material for piston and for piston rod packings, valves, and parts of pump. Page 335.
2. Describe a double, vertical, single acting air pump, having inverted steam cylinders. Page 184.
3. Describe the evaporators and distillers in common use in the navy and name their attachments. Page 356.
4. Describe a modern feed water heater. Page 352.
5. Describe briefly the steam capstans now fitted to vessels in the navy. Page 371.
6. Describe the Allen dense air ice machine, together with its operation. Page 388.
7. Describe the main circulating pump usually used on board ship and name its usual connections. Page 201.
8. Describe any steam, hydraulic, or other power, steering gear, with which you may be familiar. Page 363.
9. Give a list of all the auxiliary machinery ordinarily fitted on board a modern battleship, and give the functions of each. Chapter XIV.
10. Describe the hull drainage system of a modern battleship. Page 285.

SUBJECT NO. 2.—OPERATION AND MAINTENANCE.

1. Describe the duties of the officer in charge of a steaming watch on board a modern man-of-war. Chapter XVI.
2. Describe the duties of an officer having the day's duty in port in the engineer department of a modern man-of-war. Chapter XVII.
3. You are on duty as senior engineer officer of a vessel lying at anchor with steam on one main boiler only for auxiliary purposes, other boilers ready for starting fires. You receive orders from your commanding officer to get up steam on all boilers and be ready to get away in ten hours from the time when the order was given. Describe tersely what steps you would take to carry out this order as regards boilers, engines, and auxiliary machinery. Page 429.
4. As senior engineer officer of a vessel, upon arrival in port you receive orders from your commanding officer to secure the engines, allow fires to die out under all but one of the boilers, one boiler being reserved for auxiliary purposes; what would you do to carry the order into effect? Page 41.
5. Upon arrival in port what routine work must always be carried out in order to put the engineer's department in condition for further sea service, and for the proper care of the machinery? Page 458.
6. In an inspection of the bilges of a vessel throughout the engineer's department, what portion must be particularly looked after for evidence of corrosion?
7. When using an evaporator for distilling water for the ship's use, the water becomes oily, where would you expect to find the cause of the trouble and what would you do to correct it? Page 358.
8. Describe, briefly, how you would measure the setting of a main steam distribution valve of an engine, together with method of finding the engine centers. Page 90.
9. What are the duties of the senior engineer officer of a vessel when the vessel is placed in dry dock? Page 466.
10. Describe method of fitting a pair of spare crank pin brasses which have not previously been fitted to the pin. Page 462.
11. How would you rebabbitt a pair of crank pin brasses? Page 462.
12. The intermediate crank pin and crosshead brasses show a tendency to heat. How would you relieve the pressure on these parts? Page 454.
13. How do you gauge the amount of clearance between a journal and its brasses? Page 461.
14. To what deteriorations are the interior surfaces of boilers liable, and what precautions must be taken against them?
15. The density of the water in a vessel's boiler is increasing rapidly. Where would you expect to find the trouble? Page 464.
16. Knowledge of regulations concerning engineer department. Appendix A.

DRAWINGS.

1. The candidate will be given a blue print or drawing and required to write out a description of the piece of machinery shown, giving the functions of the various features of the machine, and the materials of which the various parts are made, as shown by the hatching of the sections.

2. The candidate will be required to make a working sketch in pencil of some simple piece of machinery. Proper proportions will not be required, but sufficient measurements must be given on the sketch to enable a workman to make a duplicate piece without having to apply his rule to the drawing.

I

The following comprehensive questions in reciprocating engineering are included for reference; they were formerly used in examinations for promotion of engineer officers in the navy.

BOILERS.

1. Give brief description of a boiler, explaining the functions of the different parts.

2. Name the different classes of shell boilers commonly used in modern war vessels and describe one (to be selected by board).

3. Name the different tubulous or coil boilers that are generally considered best and describe one (to be selected by board).

4. Give the relative advantages and disadvantages of shell boilers and coil or tubulous boilers.

5. Give brief explanation of different methods of forced draft and state its advantages and disadvantages; what, if any, bad effects it has on boiler; air pressure carried with different classes of boilers; amount of coal burned per square foot of grate surface with different air pressures.

6. Name attachments to boilers and describe any three (to be selected by board).

7. Name the principal causes of the deterioration of boilers and the means of prevention.

8. Name methods of increasing circulation in boilers and describe one with which candidate is familiar.

ENGINES.

1. Describe briefly the engines used in modern war vessels as regards their position and arrangement of cylinders, explaining the conditions that lead to the selection of one type rather than another.

2. Describe briefly the types of engines used in modern war vessels from the standpoint of the way in which the steam is expanded, and note

any changes in arrangement of valves, receivers, etc., consequent on increase of the number of stages in the expansion.

3. Name the attachments to the engines of modern war vessels and describe three (to be selected by board).

4. Explain the means employed to secure powerful engines of comparatively light weight.

VALVES AND VALVE GEARS.

1. Name the valves used with the engines of modern war vessels and describe the piston valve and one other. Explain the advantages and disadvantages of the two valves described.

2. Name the materials of which valves are made and the liners and false faces to valve seats, explaining reasons for the choice of materials.

3. Name the different kinds of valve gears commonly used with the engines of modern war vessels and describe the Stephenson link and one of the radial valve gears.

4. State briefly the advantages claimed for the radial valve gears most frequently used and state your opinion, with reasons, for the truth or falsity of their claims.

5. Explain the provision made for varying the degree of expansion employed with modern engines using modern valve gears, and what, if any difficulty, from a practical point of view, there is in using high rates of expansion; also what provision is made to enable the engines to be backed at full power while using a considerable degree of expansion.

6. State any devices with which you are acquainted for reducing wear on the valve gear and for doing the actual work of moving the valves, leaving the eccentric or equivalent device only the work of determining the phases of the valve movement. Describe one of these.

CONDENSERS AND PUMPS.

1. Describe the principal features of the condensers used on modern war vessels, including the material of which made, method of packing the tubes, provision for proper distribution of the steam, etc.

2. Name the attachments to a modern condenser, including the various pipe nozzles.

3. Describe briefly the air and circulating pumps commonly used with the engines of modern war vessels and give the advantages due to use of independent pumps; also the advantages and disadvantages of combined air and circulating pumps.

4. Describe the type of boiler feed pump most often used and tell why this type is preferred.

5. Name the steam pumps most often used as auxiliaries on board ship and describe one of them.

6. Explain how the pistons and rods of air, circulating, feed, and auxiliary pumps are packed, both for low and high pressures, and for hot and

cold water. Tell the material of which the valves of the water cylinders are made for different pressures and temperatures.

7. Name any derangements to which the pumps in previous question are liable, it being assumed that they are in good order.

AUXILIARY MACHINERY.

1. Name the different types of reversing engines commonly used with modern machinery and describe one.

2. Describe the evaporators and distillers in common use on war vessels and name the attachments. Give approximate output of an evaporator of given size, when used for making drinking water, and when used for supplying deficiency of feed.

3. Describe any good ash hoist in common use with which you are familiar.

4. Describe briefly the steam capstans and windlasses not fitted to war vessels.

5. Describe briefly any good steam or hydraulic steering gear with which you are familiar.

6. Describe any good engine-room telegraph fitted for repeating the signal.

7. Name the different methods of fitting turret-turning machinery and describe one with which you are familiar.

8. Explain the accumulator and regulator used with hydraulic machinery.

9. Name the different types of refrigerating machines used on board ship and describe any one with which you are familiar.

10. Describe any efficient governor used with marine engines.

PRACTICAL BUILDING AND REPAIRING MACHINERY.

1. Describe briefly the steps in building a marine boiler, including such items as preparing the plates, laying out and trimming to size, flanging, annealing, fitting furnace and tubes, etc.

2. Describe briefly the machine work on a large cylinder of a marine engine after it is delivered from the foundry to fit it for erection.

3. Describe briefly the machine work on any one of the following pieces: Connecting rod with forked end; piston rod and cross head; solid crank shaft; built-up crank shaft; propeller shaft and propeller, either solid or with adjustable blades.

4. Describe briefly the operation of boring out shaft tubes and struts.

5. Give a brief explanation of the salient features of molding, including the kinds of sand used, venting, cores, securing molds, etc.

6. Describe briefly the process of molding a large cylinder, either from a pattern or by "sweeping up" and in dry sand or loam. (Candidate to choose which method and material in which molded.)

7. Describe briefly the molding of a composition propeller by "sweeping up" or from pattern and in dry sand or loam. (Candidate to choose.)
8. Describe process of repairing castings by "burning" and explain limitations and precautions to be observed.
9. Explain the process of melting cast iron in a cupola, including a brief description of a modern cupola.
10. Explain the process of melting composition or brass for large and small castings.
11. Explain operation of making copper pipe and note the precautions to be observed to secure good work.
12. Describe process of making a large wrought-iron forging; also a large steel forging.
13. Describe method of repairing broken parts of machinery on board ship. (The board to select two cases and state the circumstances.)
14. Describe the process of fitting white metal in a bearing and of repairing it if melted out by hot journal.
15. Describe process of fitting lignumvitæ bearings, giving precautions to be observed.
16. Explain briefly the salient features of pattern making in connection with marine machinery.

PROPELLERS, THEORY OF STEAM ENGINE.

1. Give the generally accepted theory of propelling instruments and its application to the screw propeller.
2. Describe the essential features of a good screw propeller and explain how far they are carried out in practice.
3. Describe briefly the types of screws commonly fitted to modern war vessels.
4. Explain the advantages and disadvantages of twin and triple screws.
5. Explain the advantages of high speed in marine engines and tell how it is secured.
6. Explain the influence of inertia of reciprocating parts of engines on steadiness of running, and also influence of number and position of cranks on uniformity of crank effort. Explain method of laying out a curve of crank effort.
7. Explain the effect of multiple cylinder engines on economy in use of steam and give approximate figures for steam and coal per I. H. P. in different types of engines.
8. Give the first and the second laws of thermodynamics. Give Carnot's function and explain its use and bearing on the performance of engines in practice. Explain the advantages and disadvantages of using steam of high pressure.
9. Explain the limitations to useful expansion. Show how to compute amount of water used from an indicator diagram and explain why the

amount thus calculated is not the same as that found by actual measurement of feed.

10. Explain what is meant by superheated steam and the advantages and disadvantages of its use.

11. Explain briefly why steam is a better vehicle for heat engines than other vapors with less latent heat and lower boiling point.

12. Explain briefly the essential features of a correct reducing motion for steam-engine indicators and sketch one form.

13. Explain briefly the construction of the steam-engine indicator; demonstrate the degree of accuracy of the pencil movement of any one, and explain what difference in the selection of the piston and drum springs is caused by difference in speed of engine.

14. Explain the standardization of indicators and the necessity for it.

15. Explain the correction of indicator cards by the result obtained from standardization of the indicators and tell how it is done practically for a large number of cards, as in the case of a trial trip.

16. Draw a good practical card, as obtained from an engine in good condition, and show by dotted lines the result of defect in adjustment or fitting of the valve and other parts.

17. Explain briefly the composition of coal. Show how to calculate the thermal value of coal from its percentage composition. Explain briefly chimney draft and the effect of proportions of chimney. Tell the bearing of forced draught on the economy of combustion. Calculate temperature of furnace from data given by board and tell how it is determined practically.

18. Discuss briefly the economic and practical advantages and disadvantages of using liquid fuel instead of coal on war vessels.

STRENGTH OF MATERIALS.

NOTE.—In working problems under this subject free use of text books and works of reference will be permitted.

1. Definition of principal terms employed, such as stress, strain, elastic limit, permanent set, fatigue, neutral axis, ultimate strength, safe working strength, factor of safety, modulus of elasticity, etc.

2. Explain the effect of prolonged application of stress and of repetition of stress through wide range; also of alternation of stresses of opposite kinds on safe working strength of a material.

3. Explain how the ultimate strength and elastic limit are determined in practice and explain what is meant by the expression "strength per square inch." Give brief description of the essential features of any good testing machine with which you are familiar.

4. Describe the effect of variations of temperature on the tensile strengths of the more important metals used in marine machinery.

5. Determine thickness of shell, thickness of furnace, and diameter of braces to heads for a boiler from data furnished by the board. Show how to determine the strength of a riveted joint, as compared with the solid sheet, and state what practical considerations modify the theoretical percentage found, as you have described.

6. Determine the proper dimensions of the piston rod and connecting rod, or the crank shaft, of an engine from data furnished by board.

7. Show how to determine which is the stronger column, a square or round one of same cross-sectional area; also, with same cross-sectional area a solid or hollow one.

8. Work a problem in strength of beams from data by board. What is the strongest shape of beam in steel or wrought iron in general use? Explain what governs the choice of I, T, angle, bulb, or Z bars for use as beams.

9. Explain how you would determine the proper size for the key or feather for securing a composition propeller on a steel shaft. Explain the limiting resistance to crushing in metals which "flow" under great stress and state the metals or class of metals which present this quality.

10. Explain briefly the effect of the shape of test piece on ultimate tensile strength and elongation and tell the general practice in respect of shape and size of test piece.

11. Give the tests, other than by machine, for the steel parts of marine machinery.

12. Explain the difference between the resistance to direct crushing and the resistance to compression and give some idea of how you would determine when one and when the other would apply.

13. Name the parts of marine machinery now commonly made of forged or rolled steel; also those commonly made of steel castings.

14. Give an approximate figure for the ultimate tensile strength of the following materials, of good quality, such as would be passed for use in marine machinery: Steel boiler plate; wrought-iron boiler plate; steel forgings; steel castings; bar iron (small sizes); wrought-iron forgings like engine shafts; ordinary navy composition; Muntz metal; naval brass; manganese bronze; phosphor bronze; copper.

METALLURGY.

1. Name the principal iron ores and describe briefly the operation of making pig iron in the blast furnace.

2. Describe the common methods of making wrought iron.

3. Explain the distinction between cast and wrought iron and steel and state what kinds of the latter are adapted to use as parts of marine machinery.

4. Give a general description of the principal methods of making structural steel and give details of the open-hearth process.

5. Explain briefly the difference between the "acid" and the "basic" methods of making steel and explain the importance of the latter. Explain the effect of the principal metalloids found in steel and state the maximum amount of each permitted in steel boiler plate.

6. Explain briefly the different methods of making ingot copper and state what impurities are likely to be found in the ingots and their effect.

7. Explain briefly the difference between brass and bronze and explain the effect of varying the proportions of the constituents both as regards strength and other characteristics.

8. Give a brief description of the so-called "strong bronzes," explaining their composition and the theory as to the effect of the special ingredients which give the distinctive names.

9. Give a brief account of the defects found in steel castings and the explanations usually given to account for them.

ELECTRICITY.

1. Describe the essential features of a dynamo electric machine and give a brief account of any standard machine in common use.

2. Explain briefly the construction of electric motors and wherein they differ from dynamos.

3. Define the electric terms in common use in connection with dynamos and motors.

4. Explain the principles of electric lighting by incandescent lamps and tell what means are employed in practice to secure a steady light, whatever number of lamps may be in use.

5. Explain the advantages of transmission of power by electricity and state under what circumstances it would be more economical than the direct application of steam power.

6. Give data of relative weight of plant to do a given amount of work (say 100 I. H. P. at point of application) on board ship when steam power is applied direct and when electric motors are to be used, it being assumed that steam is taken in each case from the auxiliary steam pipe.

7. Give a brief description of the storage battery, as now constructed, and state the advantages and disadvantages, in your judgment, of their use in connection with naval vessels or their dependencies.

8. Give precautions to be observed in the practical use of dynamos and motors and state what mishaps are common or likely to occur.

DRAWING.

The candidate will be furnished a blue print of some portion of the machinery of a naval vessel and will be required to write a description of it, telling the different materials employed and explaining the object of any special features.

DUTIES AS CHIEF ENGINEER OF A SHIP IN COMMISSION.

1. Procedure on joining a ship to ascertain that machinery is in proper condition—that is, what parts should be inspected, and how to secure the needed information in the shortest time.

2. Stationing men of engineer's force for steaming watches and for general and fire quarters; the board to give the number of engines and auxiliaries and of the boilers, with idea of general arrangements as regards subdivision, and the candidate to state the necessary number of men required and their assignment.

3. Care of machinery and precautions to be observed while it is in use and while not in use to insure its readiness for service.

4. Economy in the use of coal, oil, etc. (This will involve such consideration as number of boilers to be used for a given power, rate of combustion of coal, steam pressure and point of cut-off to be used, economical speed of ship, best methods of lubrication.)

5. List of parts subject to wear, and course to pursue to maintain efficiency, together with a statement of cases where deterioration is certain and cannot be compensated.

6. List of machine and hand tools carried on modern war vessels, with statement of new and repair work that can be performed. (Both the list of tools and of work will be in a general way, not going into much detail, the object being to see that candidate knows what work can be done on board and what must be done at shop on shore.)

7. A list of the more important stores and spare parts usually carried on modern war vessels.

8. What inspection and tests would be used in selecting coal, oil, and rubber.

DUTIES ON SHORE AT A NAVY YARD OR AS AN INSPECTOR OF MACHINERY BUILDING BY CONTRACT.

1. General survey of the machinery of a ship returned for refitting at end of a cruise; state what parts would require most careful examination and what work would generally be done to put the machinery in thorough condition.

2. State how estimates of time and cost would be made in the case of repair work. (The board to give a statement of character and amount of repairs required.)

3. Name the tools and appliances generally fitted in a modern plant for the building and repair of marine machinery of large size, giving some idea of the size of the various tools, cranes, etc.

4. State amount of work that should reasonably be expected during a working day of eight hours on any one of the following kinds of work:

Turning plain shafting.

Boring large engine cylinder.

Plain work in hydraulic flanging machine. (Thickness of plate to be given.)

Riveting in hydraulic riveter. (Thickness of plates and diameter of rivets to be given.)

Drilling rivet holes in boiler shells. (Thickness of plates, diameter of holes, and kind of drilling machine, to be given.)

Planing valve seats on large cylinders.

Drilling and tapping holes in condenser tube sheets.

5. Give general idea of method of conducting evaporative test of a steam boiler, stating special precautions necessary to insure accurate results.

6. State preparations that would be made for a full power trial of machinery of a modern war vessel; including stores and instruments that should be furnished.

7. Give general idea of work to be done in "laying up" machinery of a vessel that is to be put in ordinary.

8. Being assigned to duty as Inspector of Machinery, tell special points to be looked after in inspection of any two of the following:

Cylinders.

Condensers.

Boilers.

Shafting.

Piston and connecting rods and valve stems.

Pistons and valves.

Bed plates.

Propellers.

Boiler and condenser tubes.

Brass castings generally.

In addition to the two questions under this head, the candidate will be furnished with a blue print of some part of the machinery of a naval vessel and required to write such a specification of the same as would be drawn to accompany the contract.

PRACTICAL WORK.

1. Molding any large piece of machinery in loam, such as a cylinder, propeller, condenser, etc., from a pattern.
2. Making pattern for a cylinder or other complicated piece of machinery.
3. Making propeller pattern.
4. Molding a propeller by sweeping up.
5. Machine work necessary to put a cylinder casting, as received from foundry, in condition for going in ship.
6. Machine work on a built-up propeller.
7. Machine work on a crank shaft, connecting rod, or piston rod.
8. Manufacture of any large forging of steel or iron.
9. General description of work of building a large boiler.
10. General description of running lines in ship and erecting engines in place.
11. Boring out and fitting stern tubes and shaft struts.
12. Details of building a condenser.
13. Outline of fitting and erecting steam and exhaust piping, with statement of precautions to be observed.

STRENGTH OF MATERIALS, METALLURGY, SHIPBUILDING.

(The use of works of reference will be permitted in solving problems.)

1. Work out thickness of shell, detail of riveting, thickness of furnaces, diameter of braces in steam space, size and spacing of stays for a cylindrical boiler of given diameter and steam pressure.
2. Design a crank shaft for engines of a given size from data furnished.
3. Design piston rod and connecting rod for given engine.
4. Work out principal dimensions for a cylinder of given engine, including steam ports.
5. Give brief outline of process of making cast iron, explaining how different qualities are produced.
6. Give brief outline of common method of making wrought iron bars or plates.
7. Describe briefly the principal methods of making steel used in machinery and structural work.
8. Explain in a general way the manufacture of various kinds of brasses or bronzes, explaining how the proportions of ingredients affect the principal qualities such as strength, hardness, brittleness, etc.
9. Explain in a general way the manufacture of the so-called "white metals" or "anti-friction" metals, the way in which they are used, and the reasons for their use.

10. Explain the methods of framing used in war vessels constructed of steel, particularly in the space to be occupied by machinery.
11. Explain water-tight subdivision on war vessels.
12. Explain the framing and working of plates at the stern post and where the shaft tubes are attached.
13. Explain the drainage system and its connection to the engine-room pumps.
14. Explain the precautions to be taken in attaching valves to hull to maintain the strength and prevent corrosion.
15. Explain the fitting of decks and framing of hatches near the machinery compartments.
16. Explain briefly center of gravity, center of buoyancy and meta-center. Discuss briefly the considerations that affect stability and stiffness.

ECONOMICS OF MACHINERY.

1. Explain briefly the advantages and disadvantages of high steam pressures and multiple expansion engines both for marine machinery in general and for war vessels.
2. Discuss briefly the advantages and disadvantages of the various styles of valve gear and valves in general use, and of those which are used in special cases where saving of weight and space is important.
3. Discuss briefly the considerations that govern the choice of a particular style of boiler, noting the relative merits of shell and coil boilers.
4. Discuss the methods of propulsion, giving important features to be secured in an efficient propeller, and explaining why the form of propeller now in use is most desirable; or, if you think so, why any other form would be preferable.
5. Discuss the question of the methods of securing great power in marine machinery on light weight, showing what is necessary, how far it is safe to go, and precautions in inspection of material and perfection of workmanship to be observed.

J

QUESTIONS AND PROBLEMS.

PART I.

1. Define work, energy, power. What is the measure of a horse power? What is the equivalent in B. T. U., of one horse power?

2. Find the work done in lifting 25 tons through a distance of 170 feet. Answer in foot-pounds and inch-tons.

Two and one-half tons of water falls through 15 feet in 12 minutes. Give the work done in foot-pounds per minute, and foot-tons per hour, and also expressed in horse power.

3. The diameter of the cylinder of an engine is 35 inches, stroke of piston 4 feet, mean effective pressure upon the piston 90 pounds per square inch, revolutions 125 per minute. Find the indicated horse power developed by this engine.

4. What do you understand by the *efficiency of an engine*? What would be the efficiency of an engine and boiler where the consumption of coal per hour is 2 pounds per I. H. P. Thermal value of the coal 14,500 B. T. U. per pound.

5. Define the term *Efficiency*, as used in an engineering plant or calculations. Name the various losses in the motive power of a ship, from the boiler to the propeller which effect the efficiency, and roughly give the losses of each.

6. Give an expression for the maximum efficiency of a thermally perfect engine. Suppose an engine to work between the limits of pressure of 100 pounds and 18 pounds per square inch and then between the limits of 100 pounds and 2 pounds, determine the relative efficiency. Temperatures corresponding to 100, 18, and 2 pounds, are 327.5° , 224.4° , and 126.2° respectively.

7. Suppose the expansion curve in each of the above cases be, $pv = \text{constant}$, and the steam expands to the lower pressure in each case. Compare the work done by the same amount of steam in the two methods of working.

8. Name the two principal directions in which the efficiency of an engine has been increased, and give some account of the means by which these results have been secured.

9. Show the gain due to the expansive property of steam:

1st. With a constant internal pressure and varying rates of expansion.

2d. With a constant weight of steam, increasing pressure, and rates of expansion.

10. Roughly sketch relatively to one another, the hyperbolic curve of expansion, the saturated curve and the adiabatic curve. State the steam conditions due to each expansion curve and give the equation of each. Describe the method of drawing each curve correctly.

11. What losses are inherent in the stage-expansion engine and not possessed by the simple engine? How are these losses minimized in practice? As far as the power is concerned, what effect has the high-pressure cylinder of a compound engine? Why?

12. State what is meant by isothermal, adiabatic, and saturated expansion of a gas and give the relation between volume and pressure in each case.

13. A pound of coal will generate during combustion 14,000 thermal units. Assuming that a perfect heat engine consumed one pound of coal per minute with a limit of working temperature 2440° and 60° , what would be the horse power developed?

14. Define absolute pressure, specific and relative volume of steam, ratio of expansion, saturated steam, superheated steam, and wet steam.

15. State Boyle's law connecting the pressure and volume of a gas. Deduce the general formulæ for p_m when steam expands according to Boyle's law.

16. Steam pressure absolute = 160 pounds. Cut-off = $\frac{1}{4}$ stroke. Back pressure = 10 pounds absolute. Find the mean effective pressure.

Referring to the above question, show the advantage gained by using steam expansively.

17. What do you understand by the term *liquefaction* and *initial cylinder condensation*? What produces it and why is it a serious loss of efficiency in expansive steam engines?

18. Give the different methods used for reducing liquefaction losses and thus increasing the efficiency.

What effect has the dimensions of the cylinders, the initial pressure of the steam, and the ratio of expansion on the initial condensation which takes place in the cylinder.

19. Explain briefly the advantages gained by compounding the cylinders of a steam engine, or in other words compare the advantages of a triple-expansion engine with a three-cylinder engine of the same power, each of whose cylinders takes steam directly from the boilers.

20. What is steam jacketing? What is the object in fitting a steam jacket to a cylinder? Describe how the jacket steam is kept distinct from that used in driving the piston. Explain how the use of a steam jacket diminishes the loss of efficiency in an engine despite the fact that steam is being constantly condensed in the jacket to keep the cylinder hot.

21. Upon what does the success of steam jacketing depend? What should be the temperature of the jacket steam compared with the cylinder steam? What is the course of the jacket steam in triple-expansion engines?

22. What are the advantages of using superheated steam? How is it accomplished? What is the difference between superheated steam and saturated steam?

23. Show clearly how compounding the cylinders as in a stage expansion engine has such an important effect in reducing the losses from cylinder condensation. In what two ways is the economy of a steam engine expressed? State the economy in both terms which may be expected from a high-grade triple-expansion engine.

24. Discuss the advantages and disadvantages in the use of high-pressure steam.

What theoretical and practical considerations limit the increase of pressure?

25. How are liquefaction losses reduced by the employment of superheated steam. What are the objections to the use of highly superheated steam with reciprocating engines?

How is steam superheated in practice? Explain how "throttling" or wire-drawing steam tends to superheat it.

26. What is *compression* or *cushioning*? How does it contribute to lessen the loss from cylinder condensation? In addition to increasing steam efficiency, what important mechanical advantage does cushioning possess?

27. What is understood by the terms *clearance volume*, *drop*, and the loss from drop in stage-expansion engines.

28. Sketch and describe the double poppet throttle valve, and the piston throttle valve.

29. Explain clearly the following terms: steam and exhaust ports, steam lap, exhaust lap, steam lead, exhaust lead, the point of admission of steam, the point of cut-off, release, and point of compression. When is the slide valve said to be in mid-position?

Describe the difference between an inside and outside slide valve.

30. Sketch and describe an eccentric and rod and define all its various parts. What is the *eccentricity* or *eccentric radius*?

What is a simple harmonic motion? Analyze the movement with reference to the motion given by the extremity of the eccentric rod to the valve.

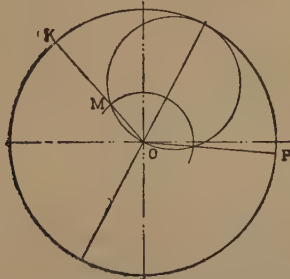
31. Show by means of outline sketches the relative position of the piston and slide valve at the four principal operations of the valve. Also sketch the valve in mid-position. Mark in the sketches the steam and exhaust lap of the valve and the lead of the valve.

32. Define the term lap and lead, as applied to a slide valve and explain their object.

What is the angular advance of the eccentric and why is it necessary?

With an outside valve and direct connection, what is the position of the eccentric with reference to the crank? With an inside valve, what is this position?

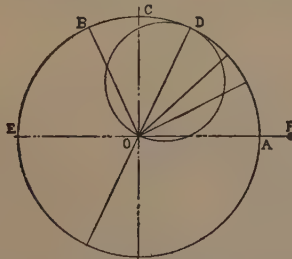
33. The port opening = distance of the valve from its mid-position minus the steam lap. Show clearly the truth of this statement by a sketch.



One of the first principles of the Zeuner diagram is as follows:

If a circle be drawn on the diagram with a radius OM = steam lap and radials OK and OP be drawn passing through the intersection of this lap circle with the valve circle, these two radials OK and OP will give the crank positions for the point of admission and the point of cut-off respectively.

Prove this statement thoroughly and conclusively.



34. A slide valve has a travel of 5 inches. Outside lap = $\frac{1}{4}$ inch and lead $\frac{1}{4}$ inch. What is the eccentricity and angular advance of the eccentric.

35. What is the object in the employment of a valve diagram? OA = throw of eccentric, OF = crank on dead center. OB = position of eccentric, COB = angle of advance. If a circle be drawn on the

line OD where the angle $COD = COB$, then any radial line from the center O and intercepted by the circumference of this circle gives the distance the valve has moved from its mid-position for the crank position corresponding to the radial line. Prove the above statement.

36. Prove the following statements in reference to the Zeuner Valve Diagram:

(a) A perpendicular left fall from the point on the eccentric circle corresponding to the angle of advance cuts the line of dead centers at a distance from the center equal to lap plus lead.

(b) A line joining the points of admission and cut-off is tangent to the steam lap circle.

(c) If with the center at the dead point and a radius equal to the lead of the valve, a circle be drawn, it will be tangent to the line joining the points of admission and cut-off. This gives a method of obtaining the lead when crank positions for admission and cut-off are given.

37. Show graphically the irregular distribution of steam on the up and down strokes of an engine, due to the effect of the connecting rod. In what ways can this inequality be adjusted? Why may the influence of the eccentric rod be neglected, while the action of the connecting rod is so apparent.

38. A complete Zeuner diagram furnishes certain information regarding the functions of a slide valve. Tabulate 12 essential items which can be obtained. Indicate those belonging to the steam side of the diagram, those belonging to the exhaust, and those common to both.

Of the above points, what are necessary to be given in order to obtain the rest, or to obtain a complete diagram?

When any of these points are given in terms of piston position, how do you determine the corresponding angular distance?

Why must one of the points in the given data be in terms of linear measurement on the diagram?

39. The half travel of a slide valve is 4 inches; the angular advance of the eccentric is 40° . Steam is cut off at $\frac{2}{3}$ stroke. Exhaust closes $\frac{3}{4}$ inch from beginning of return stroke of piston. Find by the Zeuner diagram graphically or by calculation the following in inches: (1) Steam lap, (2) Exhaust lap, (3) Steam lead. Connecting rod infinite in length.

40. From the following data construct a Zeuner valve diagram for both strokes, and from the diagram construct the theoretical indicator cards. Engine stroke = 30 inches, length of connecting rod = 4 times length of crank, travel of valve = 4 inches, lead top = $\frac{1}{4}$ inch, lead bottom = $\frac{1}{2}$ inch, top cut-off takes place when piston has traveled 22 inches, top release takes place when piston has traveled $27\frac{1}{2}$ inches,

bottom compression takes place when piston has traveled $25\frac{7}{8}$ inches, steam pressure = 20 pounds per gage and vacuum = 25 inches.

Scale of valve diagram = full size.

Scale of crank circle 2 inches = 1 foot.

Scale of indicator card, length 2 inches = 1 foot.

Pressure 1 inch = 20 pounds. For solution, see Plate 39.

41. From the above data sketch to a scale of 6 inches = 1 foot, a section of an ordinary slide valve in position on its seat when the piston is at half stroke. Width of steam ports $1\frac{7}{8}$ inch, bridges 1 inch, distance between inner edges of steam ports = $5\frac{1}{4}$ inches. Dimension the valve for top and bottom steam lap, top and bottom exhaust lap, total length of valve, and the amounts that the ports are open to steam and exhaust when the valve is in the above position (Plate 39).

42. What is the object of fitting a double ported valve in preference to a single ported slide valve?

Why are piston valves fitted to some cylinders and flat slide valves to others?

43. The low-pressure valve of a triple-expansion engine is to be fitted with a double ported slide valve—stroke of engine = 20 inches. Lead (top) = $\frac{1}{8}$ inch. Steam cut-off (top and bottom) = $\frac{3}{4}$ stroke. Exhaust opens (top and bottom) at 2 inches from end of stroke. Travel of valve = 4 inches. By means of the Zeuner diagram, find the angle of advance, steam and exhaust lap (top and bottom), and the point at which compression commences (top and bottom). Length of connecting rod = 4 times the crank (Plate 40).

44. With the following additional data make a sketch of the above valve in its mid-position on its seat.

Length of steam port = $1\frac{1}{2}$ inch. Length of exhaust port = 4 inches.

Length of bridge between steam ports = 5 inches and between exhaust ports and steam ports the length of the bridge is $1\frac{1}{2}$ inch.

Length of the bridge in the valve = 1 inch and the thickness of the metal of the valve = $\frac{1}{2}$ inch. Scale 3 inches = 1 foot (Plate 40).

45. Sketch and describe the relief arrangements such as balance rings fitted to a large slide valve.

46. What are the advantages gained by the use of the piston type of slide valve? Sketch and describe the construction of the hollow type of the piston valve. When is this type of piston valve used?

47. Sketch a longitudinal section of the solid type of piston valve with the ports and casing, and describe its action. Show a development of the valve chest lining. How is the correct distance maintained between the two discs of the valve? Of what material are the valve chest linings made?

48. How are piston valves made steam tight? Of what material are the devices made? How are they manufactured, fitted, and kept in place? Give complete details.

49. Explain the arrangement of balance pistons in general and give their use. Sketch and describe Lovekin's assistant cylinder and give its advantages.

50. Prove that with an eccentric valve motion, the point at which compression commences must vary with the rate of expansion.

51. Stroke of engine = 3 feet. Length of connecting rod = 6 feet. Steam is cut off when the crank makes angles of 60° from the line of dead centers. What is the ratio of expansion in the up and down strokes?

52. With an inside valve, would the cut-off be earlier or later when the eccentric is advanced on the shaft?

What effect would this have on the lead of the valve?

53. State in detail the method of setting the distributing valve of an engine (flat slide or piston slide), there being no marks on the engine to assist you. In placing the engine upon the center for the purpose, state clearly what you would do or direct to be done. Illustrate your description by all necessary sketches. Generalities not desired, but the proper practical methods required.

54. Describe the method of reversing the direction of motion of an engine by the "loose or shifting eccentric." Illustrate by sketch.

55. Describe the action of the Stephenson link motion and illustrate your description by a line sketch of the gear. Has the link motion any other function besides that of reversing the motion? Describe the action?

56. In the link motion the terms "open and crossed rods" are used. Define the term for both outside and inside valves. What do you understand by the terms "linking up" or "linking in"? In general what is the effect of linking in on the *eccentricity, travel of valve, lead, cut-off, and compression*? Explain by sketches the method of fixing the length and direction of the *equivalent eccentric* when linking in.

57. What is the general effect on the lead and point of compression by increasing the rate of expansion with Stephenson's link motion. (1) With open rods, (2) With crossed rods? Illustrate your answer by means of Zeuner's diagrams, the angle of advance of the equivalent eccentric corresponding to the different rates of expansion being found by geometrical method. Select your own dimensions for the gear.

58. What do you mean by full gear and how is it obtained for backing when the engine has been going ahead linked in? Sketch the contrivance used in the navy for this purpose.

59. In a link motion, what governs the radius of the link? Is there any difference if the link is to be used simply as a reversing gear and not as an expansion gear?

60. Sketch and describe the action of the steam reversing gear as applied to vessels of the U. S. Navy.

61. What do you understand by the term "multiple or stage expansion engines"? What is the total ratio of expansion in such an engine? Explain the terms double-triple and quadruple-expansion engines.

62. Make a line sketch of the Joy valve gear, describe its action, the advantages and disadvantages of the gear, how the engines are reversed, and the point of cut-off varied.

63. Explain the Wolff type and the receiver type of compound engines. Of what space does the receiver consist? In a triple-expansion engine, what is the first receiver and the second receiver? What governs the size of the receiver? What is the effect: (1) of too small a receiver, (2) receiver too large?

64. Explain the effect of the cut-off on the different cylinders in the distribution of work in a stage-expansion engine and explain how the work of the different cylinders may be nearly equalized by the cut-offs. Illustrate by indicator diagrams.

65. The power of a stage-expansion engine working at any given ratio of expansion is not affected by the size of the H. P. cylinder. Explain the truth of this statement.

66. The expansion valve on the L. P. cylinder of a stage-expansion engine does not affect the total power developed by the engine. Explain this statement.

67. Why is it better to carry out a large expansion of steam in stages as in the three-stage-expansion engine, rather than in one stage, as in the simple-expansion engine?

68. State the *mechanical* advantages of stage-expansion over simple-expansion engines.

A simple and a triple-expansion engine work at the same rates of expansion and develop the same power. What is the size of L. P. cylinder of the triple compared with the cylinder of the simple-expansion engine? State clearly the reasons for your reply.

69. Investigate the ratio of maximum to mean pressures with (1) a pair of simple-expansion engines, and (2) a compound engine of equal power, same initial pressure and ratio of expansion.

Initial pressure = 90 pounds per gage, ratio of expansion = 10. Area of each of the H. P. cylinder = A . In the compound engine

area of L. P. = $\frac{4.5}{1}$. Receiver pressure = 25 pounds.

70. Required the diameter of the H.P., I.P., and the two L.P. cylinders for an armored cruiser. Horse power of one set of twin engines=13,000. Revolutions per minute=120, stroke=4 feet. Total ratio of expansion=9.2. Cut-off in H.P. cylinder, considering clearance=.77 of stroke. Initial pressure=250 pounds per gage. Vacuum=26 inches. P varies $\frac{1}{v}$. Card factor for fast-running engines of the type and design fitted to warships=.7.

NOTE.—The card factor represents the loss due to expansion in receiver, wire drawing, compression, radiation, friction, etc., and is the

ratio of
$$\frac{\text{Expected mean pressure}}{\text{Theoretical mean pressure}}$$

71. Give roughly the fuel consumption and steam pressures of simple, double, triple, and quadruple-expansion engines as shown in marine work.

72. Sketch a longitudinal section of a steam cylinder of a large modern marine engine, showing ports, liner, piston, and its fittings in full detail. Name all the parts and state the material of which each is made.

73. Explain the use of "cylinder liners or linings." Of what material are they made? Explain clearly and illustrate by sketches how they are fitted and secured? What is the usual size of the space between the liner and cylinder body? Give the advantages of the liner.

74. Describe and illustrate by sketches the construction of a piston for a large cylinder, showing the different methods in use of making it steam tight in the cylinder. Name the materials of which the different parts are made.

75. Sketch and describe the ordinary arrangement of a stuffing box and gland. State the different materials used in packing these boxes.

76. Sketch and describe any approved form of metallic packing for piston rods of large modern naval engines.

77. Enumerate and explain the use of all the fittings and mountings attached to a large marine engine cylinder.

78. What are reducing valves, where are they applied, and for what purpose? Explain the action of any one type in general use in the navy.

79. Explain the operation of the Foster Pressure Regulator. What is the principle of its action?

80. Explain the operation of the Leslie Regulating Valve.

81. Prove that so long as an engine runs in one direction pressure is only exerted upon one of the slide bars.

When does the guide pressure become a maximum, and when is it zero? Write out a practical formula for maximum pressures on the guides.

82. In designing cross-head slippers and guide bars, what should be the maximum pressure per square inch allowed?

What are the principal disadvantages of making the connecting rod short relatively to the crank?

83. Find the thrust on the piston-rod guide and thence the area of the crosshead slipper for the H. P. engine of the armored cruiser in Question 70. Diameter of cylinder $38\frac{1}{2}$ inches. Stroke 48 inches. Length of connecting rod = 4 times the crank. Initial pressure per gage = 250 pounds. 1st. Receiver pressure = 122 pounds. Absolute cut-off = 0.7 stroke.

84. What is the cross-head of an engine? What two types are in use in the navy? Sketch and describe the open type of guide bearings. Of what materials are the different types made?

85. Sketch and describe the cross-head with closed type of guide bearings. Tabulate the materials of which the different types are made.

What are babbitted bearing surfaces? Describe the operation of babbitting. Give the proportions for good babbitt metal.

86. With reference to the force transmitted by the connecting rod, define the terms turning force and radial component. At what point of the stroke is the turning force equal to zero? When is it at a maximum?

Show by the principal of work that there is no loss of power in converting rectilinear into circular motion, by means of a crank and connecting rod.

87. For a ratio of $\frac{\text{connecting rod}}{\text{crank}} = X$ and a force P acting upon the piston, find an expression for the maximum turning force.

Show clearly the irregularity of the turning force on the crank pin. What effect has it upon the shafting, and regularity of rotation?

What other stresses is the connecting rod subjected to? What effect has speed of rotation upon these stresses, and in what part of the connecting rod are they at a maximum? What is done to the shape of the connecting rod to withstand this greater stress?

88. Sketch the usual type of connecting rod used in the U. S. naval vessels (2 views) with a forked end at the top and T-end at the bottom. Describe your sketch. How is the length of the connecting rod measured?

89. Sketch and describe two types of cap bolts in general use for connecting-rod brasses. How are they made of uniform strength

throughout their length? What are distance pieces and liners for brasses, and why are they employed? What is the usual ratio in the navy for the length of crank and connecting rods?

90. How are crank shafts with several cranks made for naval use? How are the sections of the shaft secured together? Of what material is the shaft made? What is the usual diameter of the axial hole?

What determines the *number* of coupling bolts in a crank shaft whose sections are interchangeable?

91. State the advantages of hollow shafts, pins, etc. Investigate the moment of resistance of a hollow shaft whose external diameter is d and the interior diameter d' , with that of a solid shaft whose diameter is D , and prove that with a given weight of metal you can produce a stronger shaft by making it hollow rather than solid.

92. Sketch a three-throw crank shaft for triple-expansion engines of the navy. Indicate and explain all the parts. Give the usual definition of a *bearing* and a *journal*. When a brass is made in two parts, where should the line of division be located?

93. Sketch a built-up crank shaft. How is it fitted up? Are all portions of the crank shaft of a triple-expansion engine subjected to the same stress? If not, why are all portions made of the same dimensions?

94. Describe the engine *bearers* or *seatings* of a large ship, also the bedplate. How is it secured? Of what material is it made, etc.? Illustrate by sketches.

95. How are the cylinders of a large vertical engine supported? Show the two types of framing used in the navy. How are the cylinders stayed or the columns stiffened? Illustrate by sketches.

96. Make a complete sketch of a crank-shaft bearing and its cap for a large engine. Indicate and name all the parts. State how the bearing is formed, and the object of the construction in this and other bearings.

97. Make a sketch of the shafting of a twin-screw vessel from the forward engine to the propeller, giving the technical names to each part, and naming the various bearings and their object.

98. Make a longitudinal section of a stern tube showing full details and note the materials of which each part is made.

99. State the material used for packing the stern tube and method of preparation.

100. Describe the arrangements for supporting the screw shafting after it leaves the ship.

Sketch and describe completely the details of a stern tube bearing.

What portion of the stern shaft is cased? Why is it done and how is the operation performed?

101. How is the after part of the hull protected from galvanic action due to the dissimilar metals employed in the under-water body? With sheathed ships, how is the steel shaft protected?

102. Sketch and describe the horse-shoe thrust block and explain clearly the method of adjusting each collar individually, and how the oil supply is arranged. What are steady bearings? Where are they applied?

103. Sketch and describe the ordinary collar thrust bearing. How is it adjusted? What is a spring bearing? Where and for what purpose is it used?

104. State what advantageous results are obtained from the use of the condenser with marine engines. Sketch and describe the action of the condenser as used in the U. S. naval vessels and name the material of which the various parts are made.

105. What is the objection to steam passing through the tubes and condensing water outside?

Explain and illustrate by sketches how the condenser tube ends are made water-tight, and what the result is if they leak? What provision is made for expansion and contraction? Of what material are the tubes made, their diameter and spacing? What prevents the tubes from sagging when of great lengths? What provisions are made for the examination of tube ends and tube surfaces? Explain the method of staying the tube plates.

106. What governs the amount of tube surface given to a condenser? Name all the different openings and attachments to a marine condenser.

For what purpose is the vacuum gage used? What is the usual vacuum obtained in a condenser? Why is it necessary to read the barometric pressure when the vacuum gage readings are taken?

107. If the vacuum gage on a condenser stands at 26 inches and the barometer reading is 30 inches; what is the actual pressure inside the condenser?

The temperature of the condenser is 100° . Barometer reading = 29.30. What is the greatest attainable vacuum under these conditions?

108. A cruiser engine indicating 5000 I.H.P. uses 17 pounds of steam per I.H.P. per hour and expands down to 5 pounds absolute (temperature equals 162.3°), what should be the amount of condensing surface required? How much injection water at 60° would be required per hour? Temperature of discharge water = 100° . Temperature of hot well = 125° .

109. State how a condenser is tested for leaks of all kinds.

State how tubes are cleaned inside and out.

State how to locate a leaky tube.

State how a leaky tube is replaced by another.

State the cause of loss of vacuum.

What is the best position of the air pump in relation to the condenser? Why?

110. Sketch the vertical air pump of the usual type for the condenser of a naval engine complete with the valves. Show the pipe connections and describe the method of action of the pump. Upon what does the efficiency of the pump depend, and consequently the vacuum that can be attained in the condenser?

111. Tabulate the advantages of independent air pumps for naval vessels.

Sketch and describe a single-acting air pump with but one set of valves.

112. Sketch and describe a double-acting air pump with delivery valve only, such as used in steam cutters and certain destroyers.

Of what materials are air-pump valves made? Which is preferable? Why? Sketch an air-pump valve on its seat with all appurtenances

113. Why must the capacity of an air pump of a surface condenser be so largely in excess of the volume of water into which the steam condenses per stroke?

114. Describe the feed *tank and filter*, method of construction, and fittings and pipes attached. Where are the feed tanks located? Why is a filter necessary and how is it cleaned? What filtering material is used?

Why is a hot well tank fitted in some vessels as well as a feed tank?

115. Sketch and describe a grease extractor. Where is it fitted? What are reserve feed tanks? Why are they employed and what pipe connections are led to them?

116. Sketch and describe a centrifugal circulating pump and its action.

Name the advantages of the independent circulating pump over the reciprocating circulating pump attached to the main engines.

117. Tabulate and show by a diagram the main circulating pump connections and indicate how you would free the bilge of water by this pump.

A marine engine of 2000 I. H. P. uses 15 pounds of steam per I. H. P. per hour. Temperature of exhaust steam = 140° F. Temperature of outboard discharge water from condenser = 100° . Temperature of injection = 70° . Temperature of hot well = 125° . How much injection water passes through the condenser per hour?

118. Sketch and describe an injection valve and show how it is fitted in a ship having a double bottom.

Tabulate the course of the steam from the boiler through a three-cylinder triple-expansion engine and back to the boiler again as feed water, naming all valves, pipes, etc., in the circuit.

119. What are the merits and disadvantages of copper, steel, and iron for steam pipes? What is the present practice in the U. S. Navy in specifying for main and auxiliary steam pipes for naval vessels? What care should be taken regarding the installation of pipes for salt water on board ship?

120. Sketch and describe (2 views) the conventional navy type of zinc box or zinc protectors for copper suction and discharge pipes.

121. Sketch a flanged joint of a steel main steam pipe, showing how the flange is secured to the pipe.

Describe the different methods of making the joint between the two flanges and state your opinion of the advantage of any particular type of joint mentioned.

122. Sketch and describe clearly an *expansion* or *slip joint* for a main steam pipe. Why is such a joint necessary and what prevents the joint from separating?

123. What is a steam engine indicator? Explain the principle of the indicator in its simplest form.

What points are revealed directly by indicator diagrams, or by reasoning or comparison, what particulars may be calculated or inferred?

124. Sketch and describe any type of modern steam indicator with inside spring for high-speed engines. State the area of piston used, the method of obtaining a right line motion for the pencil, the regulation of the pencil pressure against the paper, the barrel mechanism, and method of changing springs.

125. Sketch and describe any type of modern steam indicator with outside spring for high-speed engines.

126. Explain concisely and in detail the preparation and method of taking indicator diagrams. What care is taken of the indicator after all the diagrams have been taken? What scale of springs are usually supplied with an indicator?

127. Construct the theoretical indicator diagram complete from the following data: Boiler pressure 100 pounds per gage. Pressure at the engine 90 pounds per gage. Cut-off $\frac{1}{4}$ stroke. Vacuum 24 inches. Stroke of engine = 6 feet. Release occurs at 6 inches from end of forward stroke and exhaust valve closes at 7 inches before the end of the return stroke. Steam expands according to Boyle's law.

Letter, name, and describe each line of the diagram.

128. What effect has clearance upon the shape of the indicator diagram?

State why the vacuum shown by the diagram is generally less than that in the condenser. What is the usual difference?

Even with care in valve settings, why is it that diagrams from the two ends of the cylinder often differ very considerably in area and shape?

What is *wire drawing*, how occasioned, and what is its effect on the diagram?

129. Define the terms *piston clearance* and *clearance volume*. How is clearance expressed and represented on the indicator diagram? What are the effects of clearance? If steam is cut off at $\frac{1}{2}$ of the stroke and the clearance is $\frac{1}{10}$, what is the actual ratio of expansion?

What is *compression* or *cushioning*? What effect has it upon the clearance steam, and what mechanical advantage does it possess?

130. Show how the ratio of clearance may be determined from an indicator diagram.

Show by indicator diagrams the defective adjustment or construction due to—

Excessive angular advance,

Insufficient angular advance,

Excessive steam lap,

Insufficient steam lap,

Valve stem too short,

Valve stem too long.

131. What occasions a loop at the top of the admission line and at the exhaust end of the diagram? What effect has a leaky slide valve upon the diagram, a leaky piston? How is the effect of initial cylinder condensation and re-evaporation shown on a diagram?

How may the fluctuations of pressure in the steam pipe and steam chest be shown in a diagram?

132. Why is it necessary to take diagrams from each end of the cylinder in order to obtain the effective forward pressure?

Show by a diagram how to obtain the true effective pressure at any point of the stroke.

133. Explain the operation of computing the indicated horse power from an indicator diagram.

134. What is brake horse power? Sketch and describe the Prony brake. How is brake horse power determined?

What occasions the difference between the indicated and the brake horse power?

135. Sketch and explain the Coffin averaging instrument, and how it is employed in obtaining the mean effective pressure from a diagram.

136. From the accompanying cards and data, find the following:

The I. H. P. by the method of ordinates.

The I. H. P. by the Coffin averaging instrument.

The water or steam accounted for by the indicator per I. H. P. per hour.

137. Describe how, by means of the indicated diagram, an estimate may be made of the consumption of steam in an engine, and explain why the amount so measured is generally below the actual weight of steam used.

138. State how to ascertain if the piston and valves of an engine are steam tight.

What do you understand by the term *referred mean effective pressure*?

139. Explain exactly how you would draw the combined diagram of a triple-expansion engine when you are provided with a diagram from each cylinder.

140. Explain the method of combining diagrams so that the cushion steam is eliminated, the converted diagram showing the pressure and volume of the steam actually used or discharged from each cylinder per stroke.

141. Explain a method of standardizing indicators.

142. Explain the method of correcting indicator cards.

How is parallelism of the pencil movement of an indicator tested?

How test accuracy of the drum motion?

How test for back lash and friction?

143. What constitutes the drainage system and the pumping system of a ship?

Show that the initial flow of water through a hole x feet below the surface is about $14\sqrt{x}$ tons per hour per square foot of area of hole.

Under the above conditions, what size hole 16 feet below the water line would be sufficient to overcome the pumping capacity of a large cruiser or battleship.

144. Sketch and describe a manifold such as used in a drainage or pumping system and show how it is operated.

145. Sketch and describe the arrangement of suction and discharge manifolds, using as an example the boxes connected to the auxiliary feed pump. Indicate what valves are to be opened in taking water from the drainage system and pumping it overboard.

146. Sketch and describe a Macomb strainer and show how it is operated.

Explain how the centrifugal circulating pumps in the engine rooms may be made available in dealing with the inflow of water into the ship. What should be the location of this pump? Why? Why is the centrifugal type adopted?

147. Tabulate all the steam pumps to be found in the engineer department of a modern battleship, naming the suction and discharge connections to each.

Indicate by asterisk all pumps that can be utilized in freeing a ship of water.

148. What is the object of water-tight subdivisions?

Name the four methods by which water-tight subdivisions are obtained.

What is the collision bulkhead? The stuffing-box bulkhead?

What are the special advantages attendant upon a middle-line bulkhead in the engine and fire rooms?

149. Give the nomenclature of water-tight subdivisions for vessels of U. S. Navy.

150. Upon receiving an order to pump out a compartment, state in detail your procedure.

Upon receiving the order to fill the compartment state all that has to be done before and after carrying out the order.

151. What measures are taken for the preservation of the double-bottom compartments of a naval vessel and to maintain in an efficient condition the water-tight bulkhead and the drainage system?

What precaution should be taken against accidents previous to and during the inspection of the double-bottom compartments, the interior spaces of the boilers, and steam cylinders?

152. Describe with outline sketches hinged and sliding water-tight doors. What care should be taken regarding the rubber gaskets and precautions taken to keep the gear in good condition?

153. Describe with outline sketches the *Long-Arm* system for operating water-tight doors by hand and power (electric control).

Name all the parts of the power apparatus located at each door and describe the action of each.

154. Describe the drainage system of a battleship, showing how compartments communicate with each other and how the following compartments are drained, those at the extremities of the ship, the double bottoms, compartments above the protective deck, small store rooms below the protective deck, and coal-bunker compartments.

155. Describe the main and auxiliary drain pipes and their lead. The main drain was formerly led through the double bottom, why does it in present practice lead above the inner bottom? What connections are made to these drains and how are they operated?

156. Of what does the *flooding system* consist and how is it operated? What is the operation of the *automatic alarms*?

Describe the *fire service* and the *flushing service* as applied to naval vessels.

157. Name the admiralty coefficients for powering. State the assumptions upon which these formulas are based and their limitations.

158. What are the principal resistances offered to a vessel passing through the water?

Show how the relation "I. H. P. varies as the speed cubed" for moderate speeds is obtained.

159. A ship is designed for a speed of 20 knots with 16,500 I. H. P. and a displacement of 8150 tons. What I. H. P. would this ship require to drive it at 21 knots?

Maintaining this latter I. H. P., what amount of weight must be removed to increase the speed to 22 knots?

160. Describe the operation of ascertaining the horse power required for a proposed ship by means of Kirk's analysis.

161. Explain the method of powering a proposed ship by means of Froude's Law of Comparison. What do you understand by the term *corresponding speeds for similar ships*?

162. A model of a ship of 12,500 tons displacement is towed at the rate of 3 knots per hour. The model is constructed on a scale of $\frac{1}{4}$ inch = 1 foot. To what speed of the ship does this correspond?

A vessel of 10,000 tons has on trial an I. H. P. of 9500 at 16 knots. What would be the probable I. H. P. of a vessel of the same form, but of 1.25 times the displacement at the corresponding speed?

163. Explain the term *effective horse power* and the cause of the difference between I. H. P. and E. H. P.

What are progressive speed trials? For what purpose are they desired? How is the performance of a progressive trial shown graphically?

164. What are the value of *performance curves*?

What is *indicated thrust*? How is it obtained?

By means of a curve of indicated thrust, show how to obtain the *constant or initial friction*.

165. Define the term *propulsive coefficient* and give an approximate value for twin-screw ships. Show how this difference is accounted for.

From the above data stated, show the engine efficiency and the efficiency of the screw propeller.

166. Show the cause of the poor economy of high-powered naval vessels when running under cruising condition at $\frac{1}{8}$ or $\frac{1}{10}$ of the designed power.

Show by a diagram with power as ordinates and speed as abscissæ the distribution of power on a naval vessel?

167. A vessel steaming at the rate of 16.62 knots per hour requires 3570 I. H. P. with an expenditure of 2.3 pounds of coal per I. H. P. per hour. At 10 knots the I. H. P. developed is 557.6 with 1.8 pounds of coal per I. H. P. per hour. What distance can be covered on a coal supply of 150 tons at the maximum and at the reduced speed?

168. With a coal consumption of 75 tons per day a vessel steams 13 knots per hour. Bunker capacity = 800 tons. What is the maximum speed she could steam between two ports 2000 knots apart?

169. If a vessel expends 225 tons in steaming 1000 miles at 13 knots, how much will she require for a distance of 2000 miles at 10 knots (coal for auxiliary purposes excluded in each case)?

170. A ship develops 4000 I. H. P. and steams 13 knots on a coal consumption of 2.09 pounds per I. H. P. per hour. What is the shortest time she could make a port 1000 miles distant with 500 tons available coal? What is approximately the coal per I. H. P. per hour and the I. H. P. developed?

171. At full power the engine develops 4000 I. H. P. with a speed of 14 knots and a coal consumption of 2.5 pounds per I. H. P. per hour. At this speed 3000 knots can be covered with her coal supply. Determine the speed when developing 3000 I. H. P. and the distance she could steam at this speed, the coal consumption now being 2 pounds per I. H. P. per hour.

172. Show the method of drawing a coal consumption curve and from it obtaining the most economical speed.

Show the modification, taking into consideration the coal for auxiliary purposes. How may the exhaust from the auxiliaries be utilized instead of sending it into the condenser?

173. A ship whose bunkers hold 800 tons of coal makes a speed of 20 knots with 9800 I. H. P. and consumes 3 pounds of coal per I. H. P. per hour. How far can she steam at the rate of 12 knots per hour consuming 2 pounds of coal per I. H. P. per hour?

174. Describe the principle of action of the marine screw propeller. In what way is the theory of its action modified practically?

175. Define the following terms in relation to a screw propeller:

Length of screw.	Pitch, uniform and varying.
Diameter.	Fraction of pitch.
Projected area.	Angle of screw.
Face and back of blade	Leading edge.
Hub.	Right-hand screw.
Propeller race.	Pitch ratio.
Helicoidal area.	Speed of screw.

176. Show that tangent of angle of screw = $\frac{\text{pitch}}{\text{circumference}}$

Define slip, apparent, true, and negative slip, and show clearly how they are occasioned.

Show that for the propeller to be efficient, slip is necessary.

177. The pitch of the propeller of the destroyer Lawrence is $9\frac{1}{2}$ feet and with 350 r. p. m. a speed of 29 knots is attained, what is the slip per cent?

178. Deduce the formulæ for mean normal thrust and solve the following:

A vessel is driven by twin-screw engines. I. H. P. on each shaft = 5000. Speed of ship = 16 knots per hour. Diameter of shaft = 14 inches. Find the diameter of the thrust collars, assuming there are 11 on each shaft. Maximum pressure per square inch upon thrust collars = 50 pounds. The bearings are of the horse-shoe type and each with a bearing surface approximately 65 per cent of the area of thrust collar.

179. State the advantage of twin screws.

180. Describe the principal features of the type of screw propellers used for U. S. naval vessels and illustrate by sketches.

Why are blades made separate? Why is the boss or hub made so large? How is the blade attached to the hub and what is the object of this arrangement?

181. Describe fully and illustrate by sketches a practical method of measuring the pitch of the screw propellers of a vessel in dock.

182. In the case of a true screw, if you take that part of the blade where the inclination of the face is 45° to the axis of the shaft, then the pitch will be equal to the circumference due to that radial distance. Prove the truth of this statement.

183. Sketch a type of direct-acting or positive-motion pump such as used for main and auxiliary feed pumps on U. S. naval vessels. Describe fully the valve motion and illustrate your description by separate sketches.

184. What is the commercial method of designating pumps? What is the object of an air chamber on a force pump?

Why is an air chamber sometimes placed on the suction pipe of large pumps?

What is a duplex pump and with a line sketch explain its action?

How is the pump rod packed and what packing is used in the steam cylinder stuffing box?

185. What is a single-acting pump and a double-acting pump? Explain why it is possible to force water into a boiler against the steam pressure which is driving the pump, or into a boiler whose pressure is considerably in excess of that driving the pump.

In case a pump is in operation and from carelessness the check or discharge is not opened, what prevents an excess of pressure in the discharge chamber with the danger of its rupture?

186. Give suction and discharge connections of both the main and auxiliary feed pumps in U. S. naval vessels.

Of what materials is the water cylinder made, the pistons, valve seats, studs, valves, and springs?

How are the valve seats made? Sketch and describe the valve seat and valve complete with all its attachments.

With what material is the piston in the pump barrel packed?

187. From what depth will a pump draw water? Is there any limit? Explain the reason.

Sketch and describe the action of the ejector. What is the objection to the apparatus?

188. Sketch and describe the action of an injector. For what purpose is it used and what value does it possess?

189. What is the object in the installation of feed-water filters? What is the usual type of feed-water filters in the navy? What filtering material is used and how is it cleansed or renewed?

190. Why are auxiliary condensers necessary? Sketch and describe the type almost universally used in the U. S. Navy with combined air and circulating pump attached.

Indicate and name all openings for steam and water and by arrows show the course of the circulating water and the exhaust steam.

191. Why is it necessary to use fresh water only with modern boilers? What is the occasion of the loss of fresh water about a steam engine? What means are adopted to reduce this waste to a minimum?

Roughly what would you estimate the amount of *make-up* feed for a modern engine in good condition per 1000 I. H. P.?

192. Sketch (two views) and describe the construction and operation of the Bureau of Engineering type of evaporators usually installed on board U. S. naval vessels.

Name all mountings or attachments placed on the evaporator and indicate all steam or water openings and their use.

193. Describe the working of an evaporator under low pressure or exhaust steam. What is the advantage of generating steam at low absolute pressures in evaporators and what precautions are taken to avoid priming?

From what sources may the primary steam be taken to an evaporator and in what ways may the secondary steam be disposed of?

194. Describe a multiple-effect evaporating plant. What is the object of this arrangement? Sketch and describe an *ash ejector*.

195. Sketch and describe the *distiller* for drinking water in general use in the navy, lettering and naming all steam and water connections with the different pumps used. Name the materials of the different parts of the distiller.

196. Sketch and describe the ash-hoist engine generally found on U. S. naval vessels. Show how the gear is actuated by the differential valve.

197. Describe the installation for air-compressor machinery on board naval vessels. Sketch and describe the operation of a three-stage air compressor for naval purposes and name the attachments. How is the air stored?

198. Explain briefly the principle on which ice is made by an ice machine of the compressed-air type.

Sketch the Allen Dense Air Ice Machine. Name and explain the various parts and describe the operation of the machine.

199. Sketch the carbonic-acid refrigerating machine. Of what do the parts consist? Describe the operation of the machine.

200. On receiving orders to get up steam, the number of boilers having been stated, state clearly and concisely the procedure in the fire room to fully and safely carry out the orders and have steam on the boilers as required. State all the precautions you would take for safety and preservation of the machinery.

201. Steam having been raised on the boilers, state how it is controlled if time of starting has been delayed.

202. While the fire-room force is engaged in raising steam on the boilers, state clearly and in detail the operation to be attended to in the engine rooms. Steam having been obtained, continue the engine-room operations up to the time you report the department "ready."

Should the time of departure be delayed for several hours, what is your procedure then for the engine room?

203. Describe the following duties of a steaming watch in a fire room:

Management of the fires underway. Cleaning fires.

How to fire a furnace. Sweeping tubes.

204. Describe the routine of the watch under the water tender in the fire room, and state clearly his general duties in charge of the watch.

205. While the vessel is underway orders are received to start one or more additional boilers. State explicitly and in detail the operation until the boilers are connected up.

Similarly orders are received to disconnect one or more boilers. State all details of the operation.

206. State the details of the engine and fire-room routine for preparations for coming into port and on arrival at an anchorage.

207. State the details of the engine and fire-room management in getting underway and on getting orders full speed ahead and engines fairly underway. Adjustments, observations, records, etc.

208. While underway, what information can be obtained from the various sounds, thumps, knocks, etc.? State the normal and abnormal sounds you know of and how they are produced and any remedies that may be applied.

209. What are the duties of the engine-room watch or the routine in the engine room at sea, giving in detail the duties of the men stationed in the engine room and the precautions taken to maintain the machinery in good running condition?

210. Suppose your feed tanks show excessive feed water, what examinations would you make?

How do you renew a damaged grate bar?

211. State what you would do under the following circumstances:

A gage glass burst.

Low water or water out of sight.

Feed pumps inoperative.

212. What is priming in a boiler and what may result from priming? What are the causes of priming? How does the design of a boiler affect priming? How does bad management cause priming?

State clearly the indications of priming and how priming may be checked.

213. State the procedure in the following emergencies:

Collapse of a furnace crown.

Leaky tubes.

Hot bearings, causes of, how to cool a hot bearing, the distinction between *warm* and *hot* bearings and the application of water.

214. State the procedure if a piston rod commences to heat and if it continues until the rod is hot. What care should be taken to prevent initial warming?

Suppose the vacuum becomes impaired, what may be the cause and what examinations do you make?

215. Describe the management of water-tube boilers and what casualties are liable to occur.

216. On arrival in port and the necessary time allowed, state the examinations, inspections, and adjustments to be made about the main engines, auxiliaries, and the general routine of the engine-room force for port service in accordance with the U. S. Navy Regulations.

217. While the above work is in progress, state the work carried on in the fire room and its contained machinery, examinations, cleaning, repairs, etc., in preparation for sea service.

218. State the inspections, cleaning, repairs, and routine in regard to the coal bunkers, double bottoms, bilges, and machinery outside of the engine and boiler compartments.

219. Describe briefly but clearly the method of making the following adjustments, repairs, and tests. State what reference marks, if any, are made, and in case want of time interferes, any emergency method of working:

The adjustment of main bearings.

To refit a bearing, the white metal of which has run while hot.

To refit a bearing.

To adjust cross-head slippers.

In the adjustment of crank pin brasses and eccentric bearings after setting them up, to test the amount of clearance.

To adjust the thrust bearings.

To test a condenser.

Crank pin and cross-head brasses.

How are copper pipes repaired?

To straighten a bent piston rod.

220. Write out in detail the precautions to be observed in the engineer's department preparatory to going into dry dock.

The details of inspection and usual work to be done.

The name of all the sea openings and valves for whose examination and overhauling the engineer officer is responsible.

How measure the stern bearings, to ascertain if the shaft is down.

Precautions to be observed preparatory to going out of dock.

The ship to be lighted by her own electric plant while in dock.

221. What do the U. S. Naval Instructions require regarding the examinations, tests, care, etc., of the following:

Cylinders, receivers, steam jackets.	Zinc plates.
Indicators.	Holding-down bolts.
Piston rings, followers, and follower bolts.	Surface condensers.
Filters and grease extractors.	Air and circulating pumps.

How often must all cocks and valves be moved?

When not steaming, how often are main engines and valves moved and auxiliary machinery jacked?

222. Give the requirements of the U. S. Naval Instructions regarding the following:

The best working condition regarding point of cut-off.

The use of oils, and in what parts certain qualities are prohibited.

Care of engine-room signal apparatus.

Rubber valves. Examination of cylinders and valve chest after steaming.

Use of water on bearings.

Care and examination of pumps. Of evaporators.

223. Give the requirements of the U. S. Naval Instructions in regard to the following:

Hauling fires after steaming.	Method of correcting condition of
The use of the blowers.	water in boilers.
Care in accumulation of ashes.	How often the boilers are to be
The condition of the water in the	examined.
boilers.	

224. State briefly but concisely the U. S. Naval Instructions regarding the care and preservation of boilers.

225. Describe the method of testing boilers on board ship. What test pressure do you apply? While the test is on, what examinations do you make?

226. When a ship is ordered out of commission, explain the procedure of getting the engines and auxiliary machinery in condition for acceptance by the navy yard authorities.

PART II.

1. Describe briefly the means that have been adopted to reduce the weight of machinery of modern vessels.

2. In comparing modern engines with the older types, the former burn considerably less fuel per I. H. P. and the weight of machinery per

I. H. P. is considerably less. State the improvements which have produced these results.

3. Enumerate the valves in direct connection with the sea which are necessary in the engine and fire rooms, and state briefly the purpose for which each is fitted.

4. What particular examination is to be made and what defects are liable to have developed after having used the double bottom compartments for trimming ship?

What is the method of treatment of double bottoms for their proper preservation?

5. Describe the line of shafting from the engine to the screw propeller, with the various bearings and their object.

6. Describe the different kinds of packing used in the navy and for what boxes are the various descriptions most suitable.

7. The following are given to determine the horse power required to drive a ship at a speed of 16 knots per hour. Use three methods:

Displacement to L. W. L., 1703 tons.

Length between perpendiculars, 228 feet.

Draught, aft, 15 feet 7 inches; mean, 14 feet $2\frac{1}{2}$ inches; forward, 12 feet 10 inches.

From trial of sister ship, $C_1 = 540$, $C_2 = 176.8$.

Volume of L. P. $\div$ Volume of H. P., 5.266.

Cut-off in H. P., 0.8 of stroke.

Immersed midship section, 435.3 square feet.

8. A triple-expansion marine engine develops one I. H. P. for each 15 pounds of steam delivered per hour. Temperature of steam 372° F. Temperature of feed 130° F. Efficiency of boiler .70. Coal consumed per hour 2240 pounds. Analysis of coal $C = 89$ per cent, $H = 4$ per cent, $O = 5$ per cent, and ash $= 5.5$ per cent. Find the I. H. P. developed.

9. While steaming at sea the I. P. piston rod is bent sufficiently to cause the piston to strike. What should be done to permit steaming into port? Describe how the rod should then be repaired. Roughly estimate time and cost of the repairs.

10. What precautions are taken to keep the feed water free from grease or other impurities? How is the feed tank cared for to attain the same object?

11. While steaming the water tender reports a boiler tube leaking in a Scotch boiler, what examination should be immediately made? Finding a tube pitted what temporary repairs are made? Make a sketch of the means used for these repairs.

12. The crank-pin brasses heat. How is the fact discovered and what is done?

13. What duties do you require of the oiler preparatory to getting underway, while underway and coming to anchor?

14. What duties are required of a water tender while raising steam preparatory to getting underway, while underway, and after coming to anchor until port watches are set.

15. What do you require as to the duties of a fireman when in charge of the evaporating plant, explaining the methods of overhauling and cleaning, saturation, etc., testing distilled water and giving the necessary frequency of such operations.

16. Orders are received while steaming to use forced draught. State in detail what is done.

While running at full speed, forced draft, the boilers prime. What is the probable cause, what is the danger? While running at full speed under forced draft, the annunciator rings STOP, what is done?

17. State briefly how the following derangements are remedied:

A leaky boiler tube.

A leaky seam.

A leaky rivet.

A leaky condenser tube.

A leaky steam pipe joint.

A leaky steam stop valve.

A leaky stuffing box with metallic packing.

A leaky manhole plate joint.

A leak in main feed pipe.

A broken gage glass.

A puncture in keel condenser of steam cutter.

A broken grate bar.

18. Briefly what is done in the following emergencies:

A main cross-head pin heats.

The steam reversing gear refuses to act.

The air pump stops, engines in operation.

The vacuum fails.

A water hammer develops in main steam pipe.

The main steam pipe bursts.

The main bilge pumps fail to take water.

The engine-room floods beyond the capacity of bilge pumps.

The main feed pumps fail.

The water disappears from the boiler gage glass.

A fire reported in a coal bunker.

19. The white metal in the bottom brass of one of the main bearings has run while hot. Describe each step in overhauling the bearing, and estimate the time necessary to accomplish the work.

20. The crank-pin brasses are commencing to heat. How is the fact discovered and what is done?

While steaming the vacuum commences to fall, what are the indications and what examination is to be made at once?

One of the boiler feed pumps is reported as failing to feed. What is done, what examination is made, what is the probable derangement, and how is it rectified?

21. Sketch, describe, and explain the operation of the following:

Sight feed oil cup.

Separator for main steam pipe.

Modern steam trap.

Injector.

22. Describe the method of refilling and readjusting a set of crank-pin brasses that have been cut by overheating.

In what respect does this differ from the method of fitting a pair of crank-shaft brasses?

23. Describe in detail the method of draining the double bottom engine and boiler compartments and coal bunkers of a modern naval vessel, making all necessary line sketches for illustration.

24. State the material and grade of which the following parts are constructed in a large modern marine engine:

Main steam pipe.	Air-pump valves.
Piston.	Fire-pump valves.
Bearing surfaces of crank-pin brasses.	Engine framing.
Cylinders.	Crank shaft.
Cylinder liner.	Main engine piston-rod packing.
Propeller.	Connecting rod.
Bearing surface in stern tube.	Air-pump bucket packing.
Engine bed plate.	Condenser shell.
	Feed-pump valves.

25. Explain the function of each of the following:

Separator.	Ejector.	Hydrokineter.
Evaporator.	Reducing valve.	Sentinel valve.
Injector.	Expansion joint.	Tachometer.
Sight feed oil cup.	Steam trap.	

26. What different kinds of oil are used in the engineer's department, and for what purpose?

For what purpose in the above department is sal soda, sulphur, black lead, and silver nitrate solution used?

27. Give the installation of workshop machinery for a first-class battleship. What patterns or templates of the machinery are usually carried?

28. Describe the method of making the following tests on board ship:

Of steam and vacuum gages.

Of a boiler to assure safe limit of pressure.

Of a safety valve to ascertain for what pressure it is set.

Of a condenser to discover leaks.

Of coal, oil, and rubber for quality and acceptance.

29. At "fire quarters" underway, give the distribution of officers and men of the engineer's division and details of "securing."

30. What causes acidity in the water of a marine boiler? What injurious effect may it have and how is its acidity detected? What preventatives are employed and how are they applied?

31. Name the tools and appliances necessary to properly equip the engineer's department on board a naval vessel.

32. What precautions should be taken to prevent water hammer and how should steam pipes be fitted to lessen the danger of water hammer?

K

TABLE OF CYLINDER CLEARANCES FOR VARIOUS NAVAL ENGINES.

Vessel.	Date of design.	Diam. of cyls. in inches.			Stroke, inches.	Number and type of valves.			² Volumetric clearance, per cent of cyl. volume.			³ Linear clear- ance, inches.	
		H. P.	I. P.	L. P.		H. P.	I. P.	L. P.	H. P.	I. P.	L. P.	Top.	Bot.
Wilmington...	1893	14 $\frac{1}{2}$	22	33 $\frac{3}{4}$	18	1-piston.	2-piston.	1-slide..	23.8	22.3	9.7	1	
Illinois.....	1896	33 $\frac{1}{2}$	51	73	48	1-piston.	2-piston.	4-piston.	19.6	21.5	19.1		
Ozark.....	1898	17	26 $\frac{1}{2}$	40	24	1-piston.	2-piston.	1-slide..	23.5	18.2	13.1		
Missouri.....	1898	34 $\frac{3}{4}$	53	163	48	1-piston.	2-piston.	2-piston.	21.6	21.9	16.7		
Virginia.....	1900	35	57	166	48	1-piston.	2-piston.	2-piston.	25.1	21.4	19.5		
Charleston.....	1900	36	59 $\frac{1}{2}$	139	45	1-piston.	2-piston.	2-piston.	29.9	23.5	19.7		
Minnesota.....	1903	32 $\frac{3}{4}$	53	161	48	1-piston.	2-piston.	2-piston.	23.1	21.3	18.4		
Montana.....	1904	38 $\frac{1}{2}$	63 $\frac{1}{2}$	174	48	1-piston.	2-piston.	2-piston.	24.5	20.7	17		
South Carolina.	1906	32	52	172	48	1-piston.	2-piston.	2-piston.	16.1	12.9	11.9		
Delaware.....	1907	38 $\frac{1}{2}$	57	176	48	1-piston.	2-piston.	2-piston.	16.2	13.6	12.5		
Pennsylvania ⁵ .	1912	40 $\frac{1}{2}$	68	190 $\frac{1}{2}$	54	1-piston.	2-piston.	2-piston.	19.5	10 ⁴	11 ⁴		

¹ Two L. P. cylinders. ² Mean top and bottom. ³ All cylinders. ⁴ Designed.⁵ Design for Pennsylvania not used; turbines being installed in that vessel.

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